RESEARCH JOINT VENTURE AGREEMENT No.
INT-91558-RJVA with
UNIVERSITY OF NEW SOUTH WALES
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DEPARTMENT OF MATHEMATICS

US FOREST SERVICE–ADFA
JOINT RESEARCH VENTURE
INT–91558–RJVA

FINAL REPORT

September 1991

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CAMPBELL ACT 2600

FINAL REPORT FOR US FOREST SERVICE–ADFA JOINT RESEARCH VENTURE # INT–91558–RJVA

The aim of this Joint Research Venture was to begin construction of a second generation fire behaviour model for the use of the US Forest Service. This model is intended to be flexible enough for use by other land use agencies in the US and abroad for predicting fire behaviour. It is proposed that this model eventually replace the model developed by Rothermel (1972), which has been widely employed in the past 20 years, but which is known to predict poorly in certain situations.

The new model is being developed in 6 stages:

- Derivation of a basic model from heat transfer considerations;
- Burning laboratory fires in uniform fuels;
- Burning laboratory fires in mixtures of fuels of different sizes and moisture contents;
- Analysis of the data produced from the laboratory fires;
- Collection of data on experimental and wildland fires in the field;
- Determination of the parameters of the model from laboratory and field data.

In 1989, while we were visiting the Intermountain Fire Sciences Laboratory in Missoula for 6 months, we did preliminary work with Dick Rothermel (Project Leader of the Fire Behaviour Project) on modelling heat transfer. We also burned a series of experimental fires in fuel beds of different depths, as well as a series of fires in mixed fuel beds. The program of experimental fires was extended, and has continued through 1990 and 1991, and work has continued on the modelling. Fires were burned at IFSL, and the results sent to us at ADFA for analysis.

In February 1991 we went to IFSL to work for 3 weeks on the project. Partial funding was provided by the US Forest Service. While we were in Missoula we worked mainly on developing the heat transfer model, but also completed the series of fires in mixed fuels that we had initiated. A very vigorous research communication has carried on since then. Experimental burns are to continue throughout 1991, and into 1992. Much of the work from March to July 1991 has concentrated on developing a robust system to automate the data analysis from the output of the laboratory instrumentation.

We are to go back to Missoula at the end of October 1991 for 3 weeks to work on

experiments in the wind tunnel to investigate the distance of the convective heating in front of the main fire propagation wave, and to continue experimentation in mixed fuels. We will also do further work on developing the basic model.

We give here a progress report on the various components of the project.

1. Development of the basic model

The model is still tentative. We give our basic ideas the Appendix. Some of our theories may need modifying as we obtain the results of the laboratory experiments. Work is to continue on the theoretical modelling in October/November. We are also planning to do instrumented experiments with thermocouples embedded within the fuel to obtain more information on the convective heating, including flame bathing of the unburned fuel ahead of the fire.

2. Burning laboratory fires in uniform fuels

During our February visit we decided, in collaboration with Dick Rothermel, on the range and conditions necessary to obtain sufficient information to fit and test the proposed physical model. As more fires were burned we considered extensions of the model, and have identified extra experiments which are needed to test the model more thoroughly. To date 130 fires have been burned in pine needles and excelsior at varying depths, packing ratios and moisture contents. Techniques have been developed to obtain estimates of the following fire behaviour characteristics:

- rate of spread;
- flame depth;
- flame length;
- flame angle;
- average reaction intensity (heat given out by the fire per unit horizonal area);
- maximum reaction intensity;
- Byram's fireline intensity (heat given out by the fire per unit length of fire line);
- reaction time (time for flaming combustion to occur).

Further information on the techniques used is given in section 4. Figures 1–8 show some of the results which we have obtained so far for the excelsior fires, and Figures 9–16 give the corresponding results for the pine needle fires. This information will be used to develop our physical model, and determine the parameters of the final model.

3. Burning laboratory fires in a mixture of fuels

23 fires have been burned in fuel beds composed of 6 mm ponderosa pine sticks, or in mixtures of excelsior and sticks. We have jointly written a paper with Dick Rothermel (Catchpole, Catchpole & Rothermel) discussing the methods used and results obtained. This paper is currently in the review draft stage, and will shortly be submitted to the International Journal of Wildland Fire.

It has proved difficult to do experiments in mixed live and dead fuels. We have developed a method of combining the effects of moisture on live and dead fuels by modifying the moisture damping term in Wilson (1990). The resulting formula for moisture damping has two parameters that can be determined by fitting to field data. The method is explained in Catchpole and Catchpole (1991) which is to appear in the next issue of the International Journal of Wildland Fire.

4. Analysis of the laboratory data

We have spent much of March–August 1991 in developing the data analysis techniques. The data from the laboratory fires is collected by the Labtech system. Data files containing the following information are output:

- mass of fuel remaining on the weighing platform as the fuel is burned;
- voltages from 16 photocells;
- voltages from 7 thermocouples;
- wind tunnel windspeed, temperature and pressure;

Information on wind tunnel conditions is used to check that conditions remain stable in the tunnel during burning.

A typical massloss plot overlaid with the photocell voltage of the photocell nearest the weighing platform is shown in Figure 17. At present the thermocouple information is not being used in the data anaylysis, but it will be used in the investigation of the convective heating process. The average time that the photocell voltage is greater than certain critical levels is used to determine the time of flaming combustion, and then from the massloss trace the various intensities associated with the fire may be calculated. Complete details of the procedures and equations used are given in the Study Plan # 4401-51, 'Second-generation fire spread model', which we have been developing over the past year.

5. Collection of field data

Data is to be collected from fires in

- grassland;
- shrubland;
- coniferous forests;
- deciduous forests;
- eucalypt forests;
- logging slash.

A large data set of about 150 experimental and 10 wildfires in Australian grassland will shortly be published by CSIRO Division of Forestry and Forest Products. To this we can add published data on low-windspeed fires in grasslands in the U.S. (Sneeuwjagt & Frandsen 1977), and in the savannah in S.Africa (Van Wilgen & Wills 1988). This will give very good information on fire behaviour in grasslands.

Information on about 60 low-medium windspeed experimental fires in shrublands in the U.S., Australia, S. Africa, and Europe is available in the literature, and we have put this onto a database. Information on medium-high windspeed shrubland wildfires in Australia is at present being analysed.

The Canadian Forestry data base contains information on about 150 fires in Canadian coniferous forests for both surface and crown fires. There is also information on 30 fires in deciduous forests and 150 fires in logging slash. This information may not, however, be sufficiently detailed for our purposes. We may therefore have to review the original publications describing these fires to extract further information.

IFSL is at present building up a database from the literature of fires in coniferous and deciduous forests, and in logging slash in the U.S. We also hope to include information on fires in eucalypt forests in Australia.

6. Determination of the parameters of the model

Work on this topic will not begin until the model is developed further.

E.A. Catchpole
W.R. Catchpole
27th Sept 1991

References

Catchpole, E.A. & Catchpole, W.R. (1991) Modelling moisture damping for firespread in a mixture of live and dead fuels. *Int. J. Wildland Fire* **1**:103–108.

Catchpole, E.A., Catchpole, W.R. and Rothermel, R.C. Fire behaviour experiments in mixed fuel complexes. (In preparation – to be submitted to Int. J. Wildland Fire).

Rothermel, R.C. (1972) A mathematical model for predicting fire spread in wildland fuels. USDA For. Serv. Res. Pap. INT-115, Intermt. For. and Range Exp. Stn., Ogden, Ut., USA, 40pp.

Sneeuwjagt, R.J. & Frandsen, W.H. (1977) Behavior of experimental grass fires vs. predictions based on Rothermel's fire model. *Can. J. For. Res.*, **7**:357–367.

Van Wilgen, B.W. & Wills, A.J. (1988) Fire behaviour prediction in savanna vegetation. *S. Af. J. Wildl. Res.*, **18**:41–46.

Wilson, R.A. (1990) Reexamination of Rothermel's fire spread equations in no-wind and no-slope conditions. USDA For. Serv. Res. Pap. INT-434, Intermt. Res. Stn., Ogden, UT, USA, 13pp.

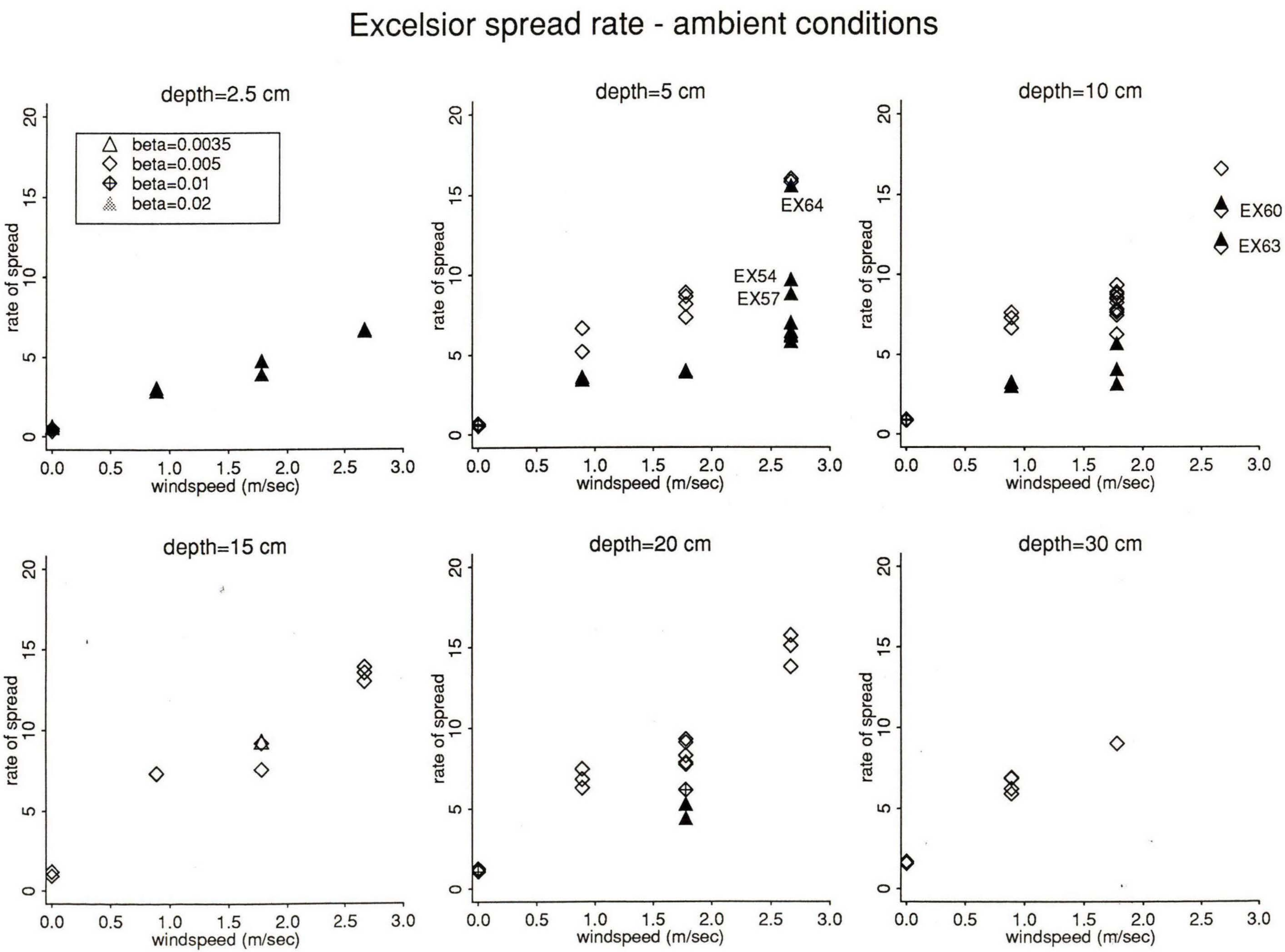
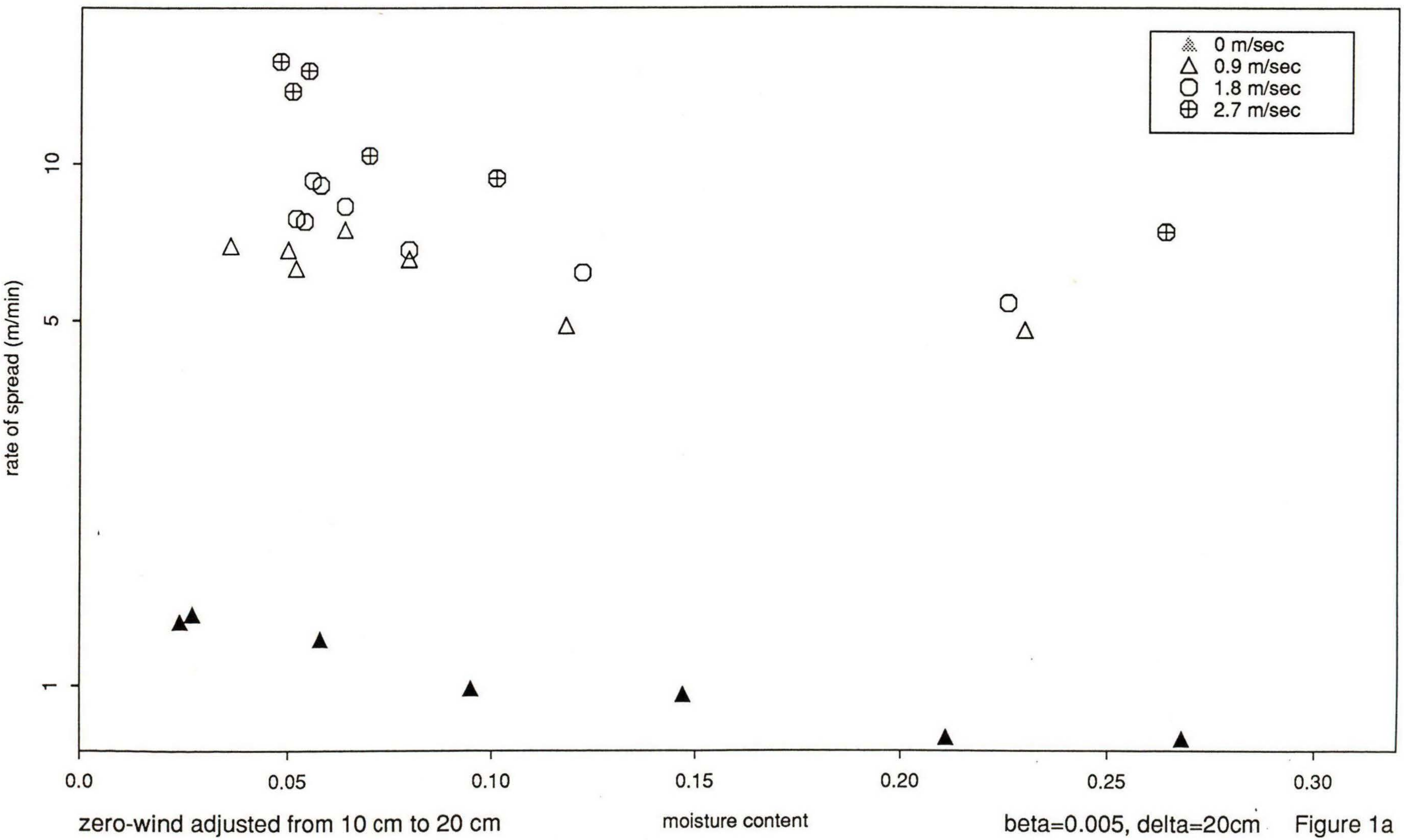

Excelsior spread rate - ambient conditions
depth=2.5 cm
beta=0.0035
beta=0.005
beta=0.01
beta=0.02
depth=5 cm
EX64
EX54
EX57
depth=10 cm
EX60
EX63
depth=15 cm
depth=20 cm
depth=30 cm
rate of spread
windspeed (m/sec)
0
5
10
15
20
0.0
0.5
1.0
1.5
2.0
2.5
3.0

Figure 1

Excelsior spread rate versus moisture content



zero-wind adjusted from 10 cm to 20 cm

beta=0.005, delta=20cm Figure 1a

Excelsior flame length - ambient conditions

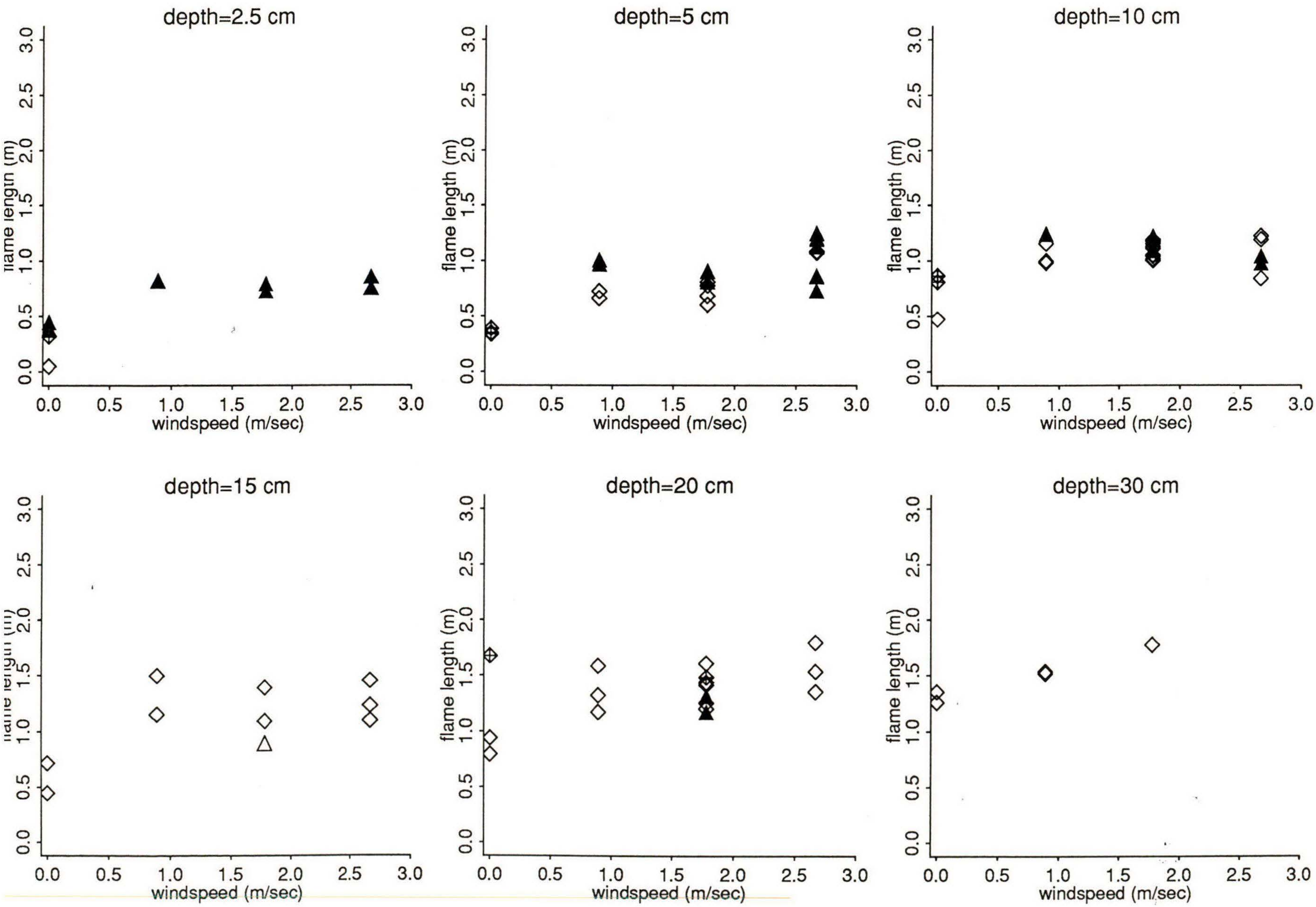
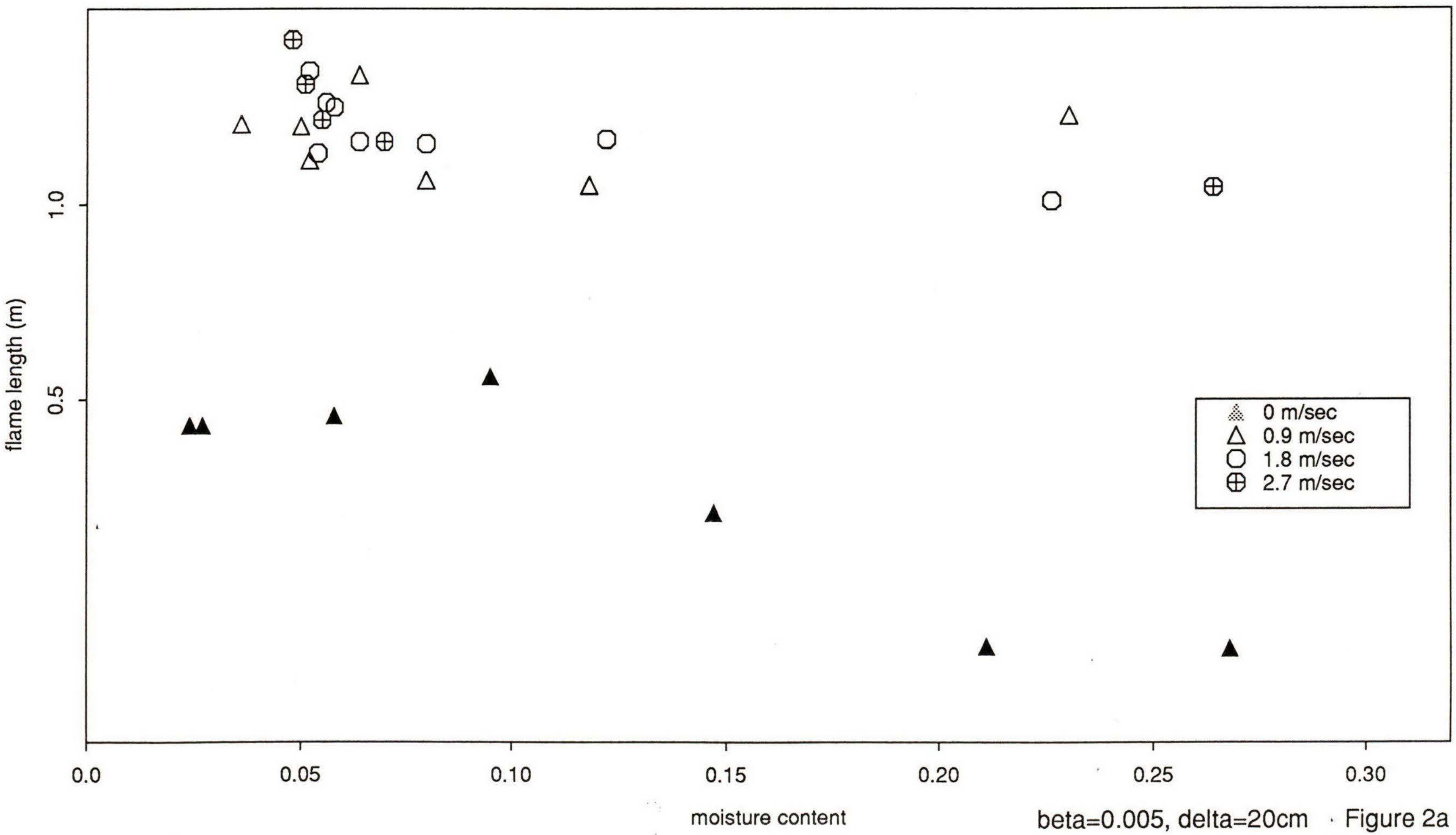


Figure 2

Excelsior flame length versus moisture content



beta=0.005, delta=20cm · Figure 2a

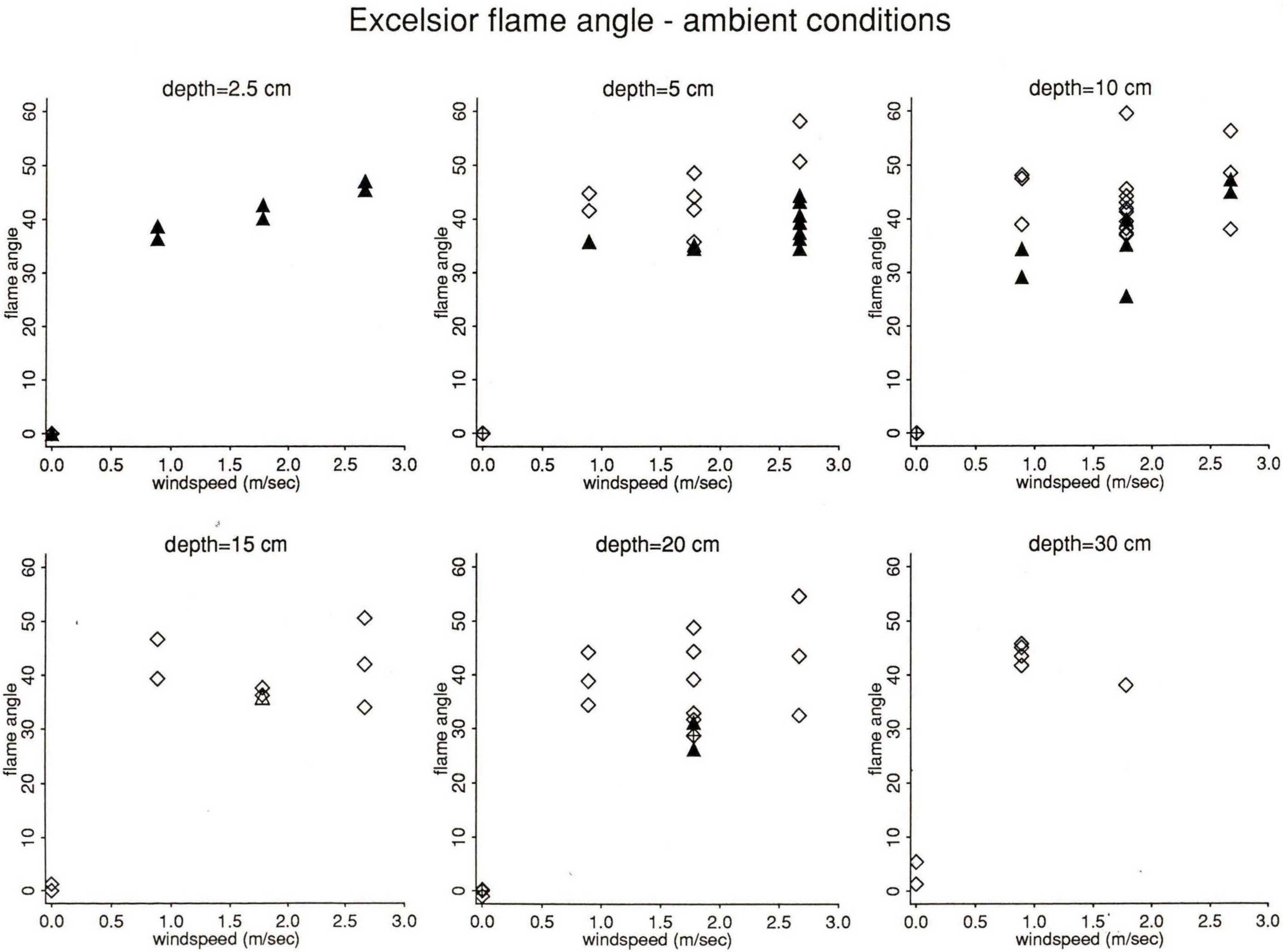
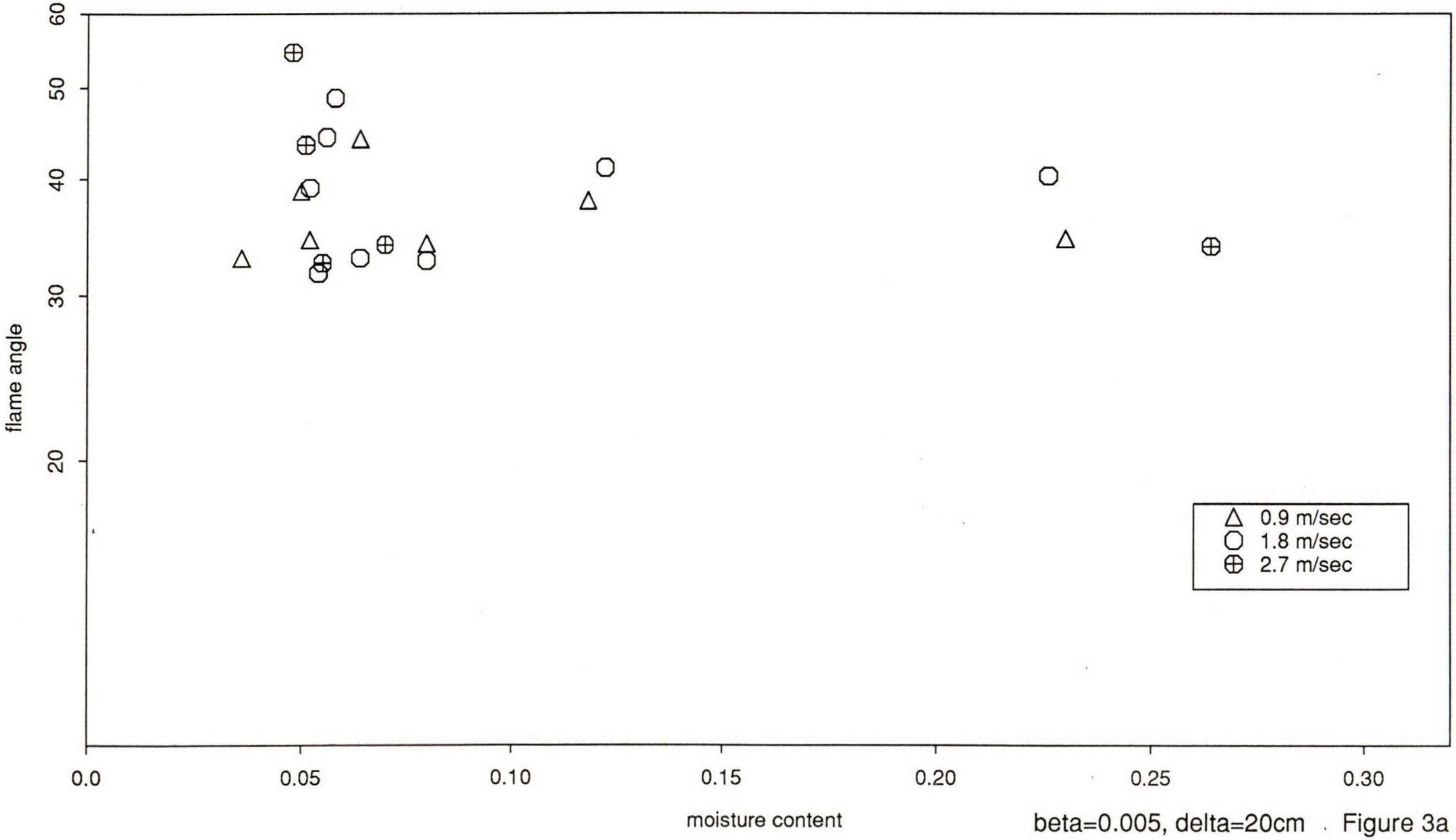
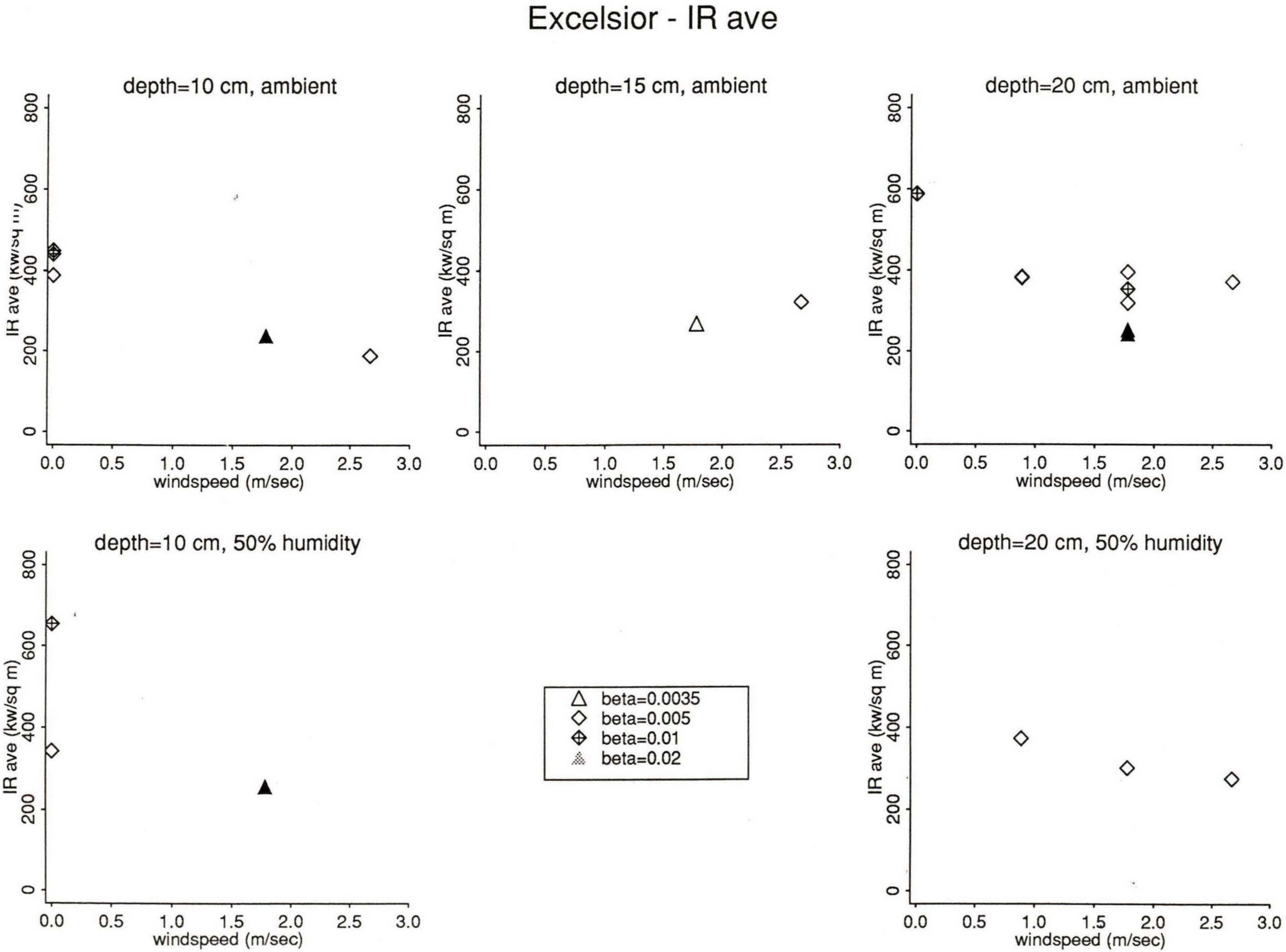

Excelsior flame angle - ambient conditions
depth=2.5 cm
depth=5 cm
depth=10 cm
depth=15 cm
depth=20 cm
depth=30 cm
flame angle
windspeed (m/sec)
0 10 20 30 40 50 60
0.0 0.5 1.0 1.5 2.0 2.5 3.0

Figure 3

Excelsior flame angle versus moisture content



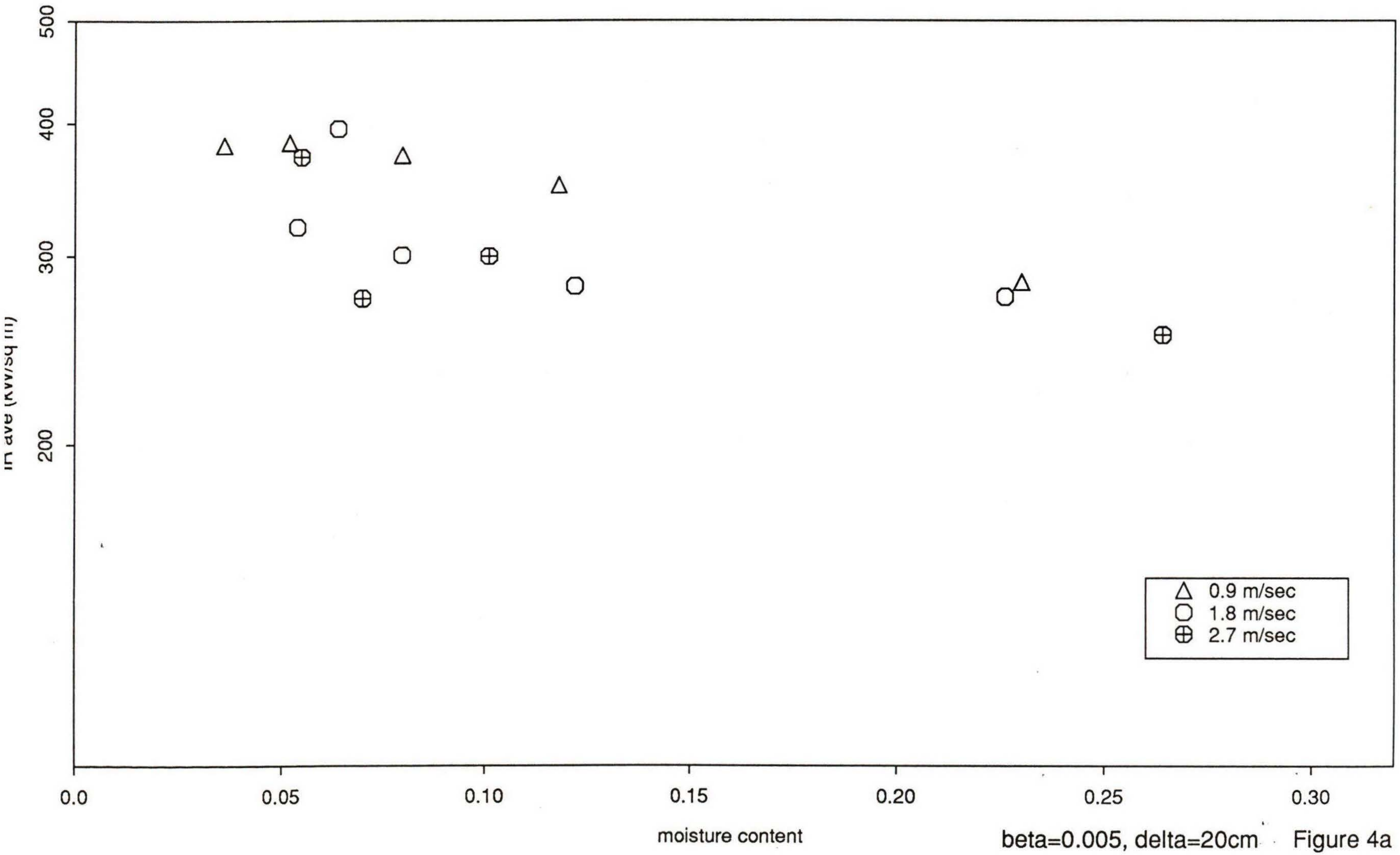
beta=0.005, delta=20cm . Figure 3a



Excelsior - IR ave
depth=10 cm, ambient
depth=15 cm, ambient
depth=20 cm, ambient
depth=10 cm, 50% humidity
depth=20 cm, 50% humidity
IR ave (kw/sq m)
windspeed (m/sec)
0
200
400
600
800
0.0
0.5
1.0
1.5
2.0
2.5
3.0
beta=0.0035
beta=0.005
beta=0.01
beta=0.02

Figure 4

Excelsior IR average versus moisture content



beta=0.005, delta=20cm Figure 4a

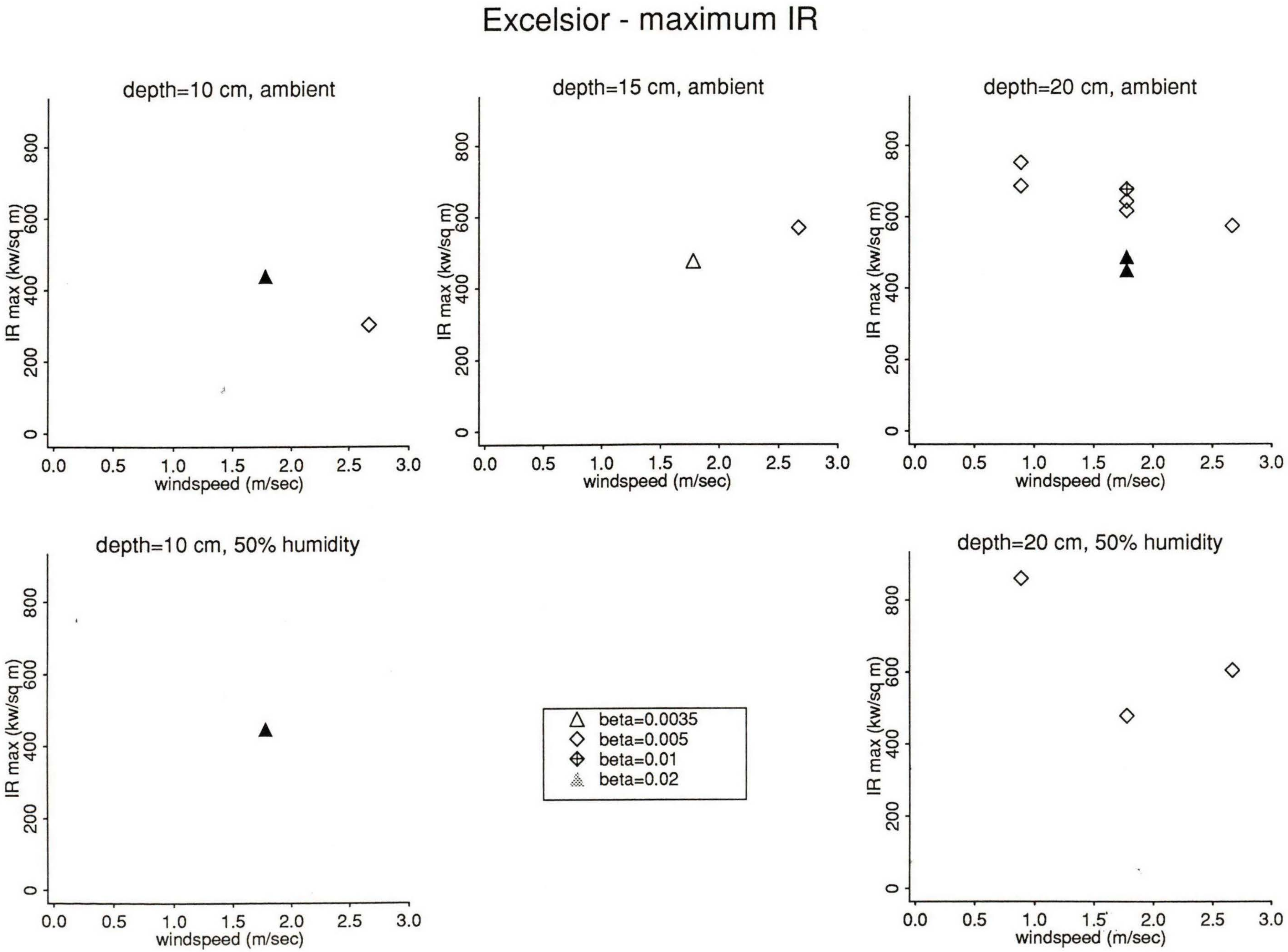

Excelsior - maximum IR
depth=10 cm, ambient
depth=15 cm, ambient
depth=20 cm, ambient
depth=10 cm, 50% humidity
depth=20 cm, 50% humidity
IR max (kw/sq m)
windspeed (m/sec)
0
200
400
600
800
0.0
0.5
1.0
1.5
2.0
2.5
3.0
beta=0.0035
beta=0.005
beta=0.01
beta=0.02

Figure 5

Excelsior - Byrams intensity

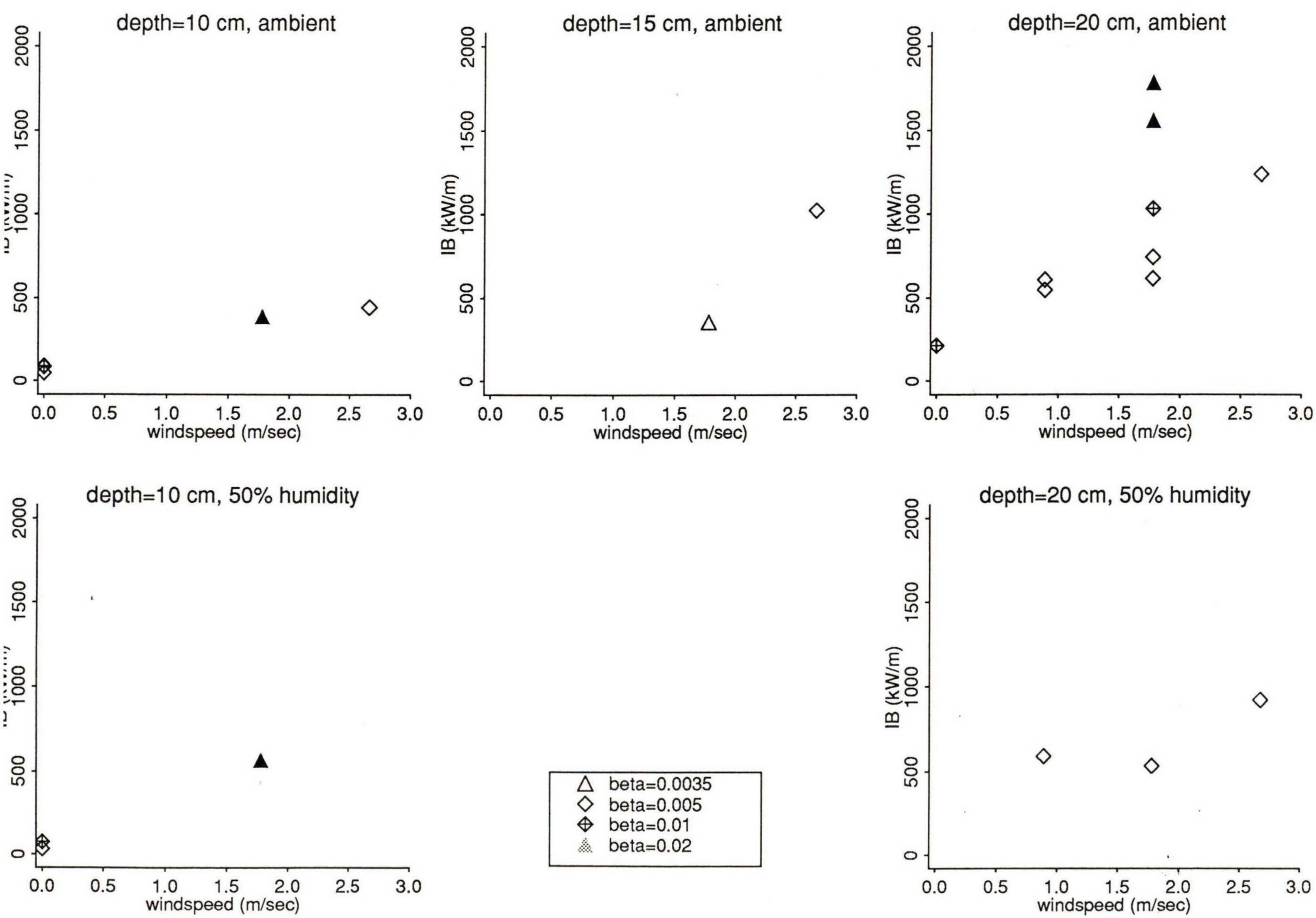
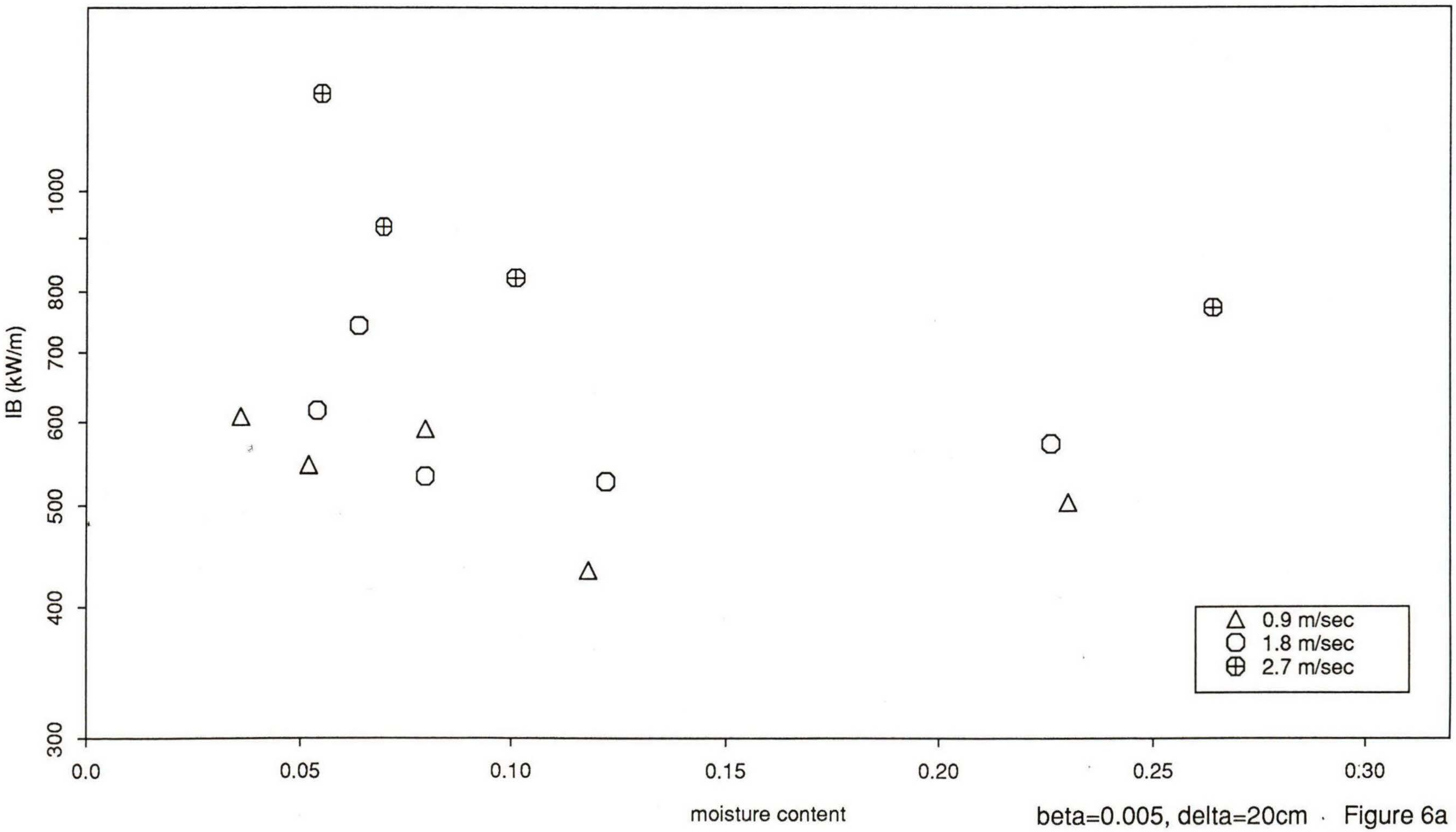


Figure 6

Excelsior -- Byrams intensity versus moisture content



beta=0.005, delta=20cm · Figure 6a

Excelsior - reaction time

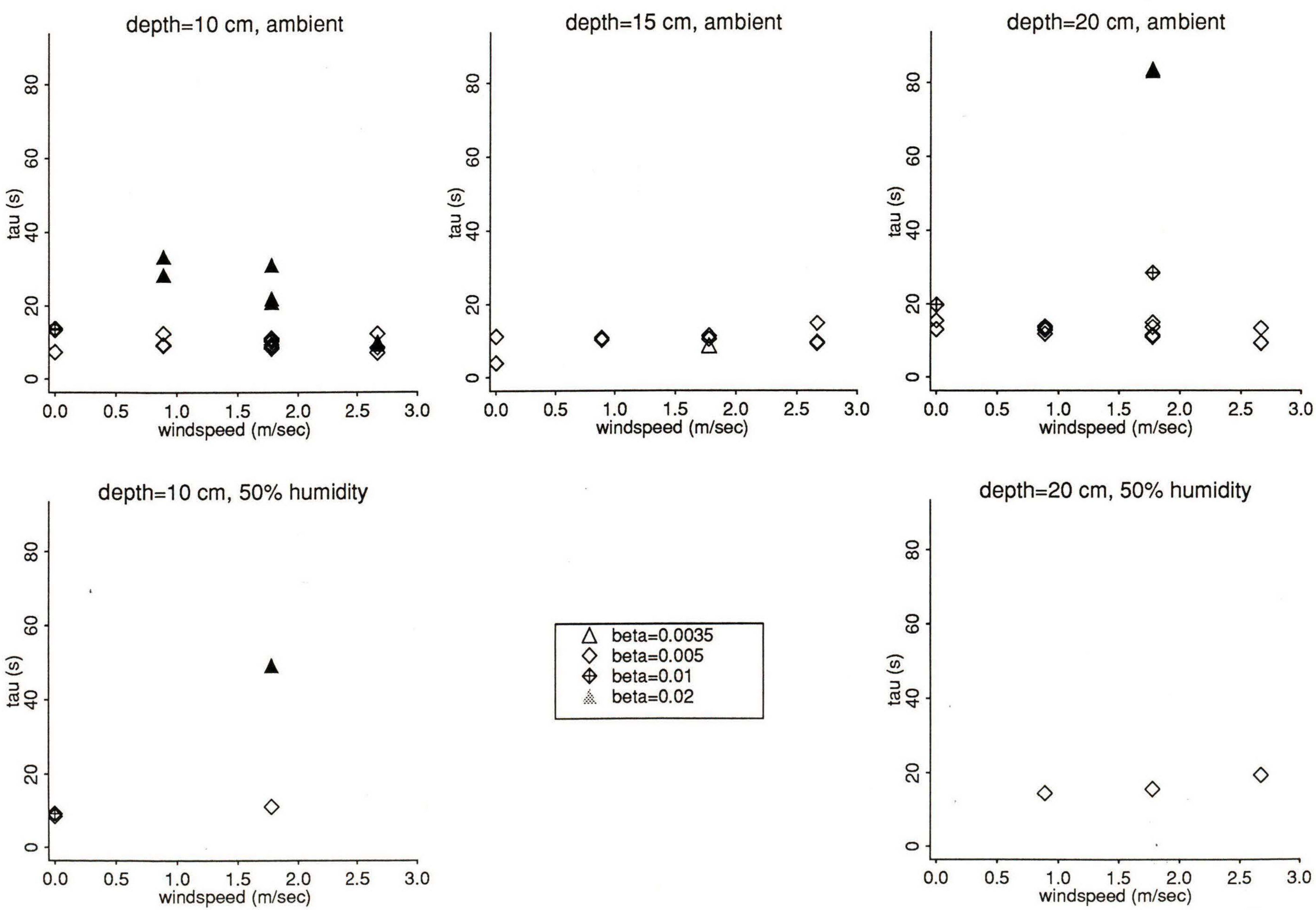
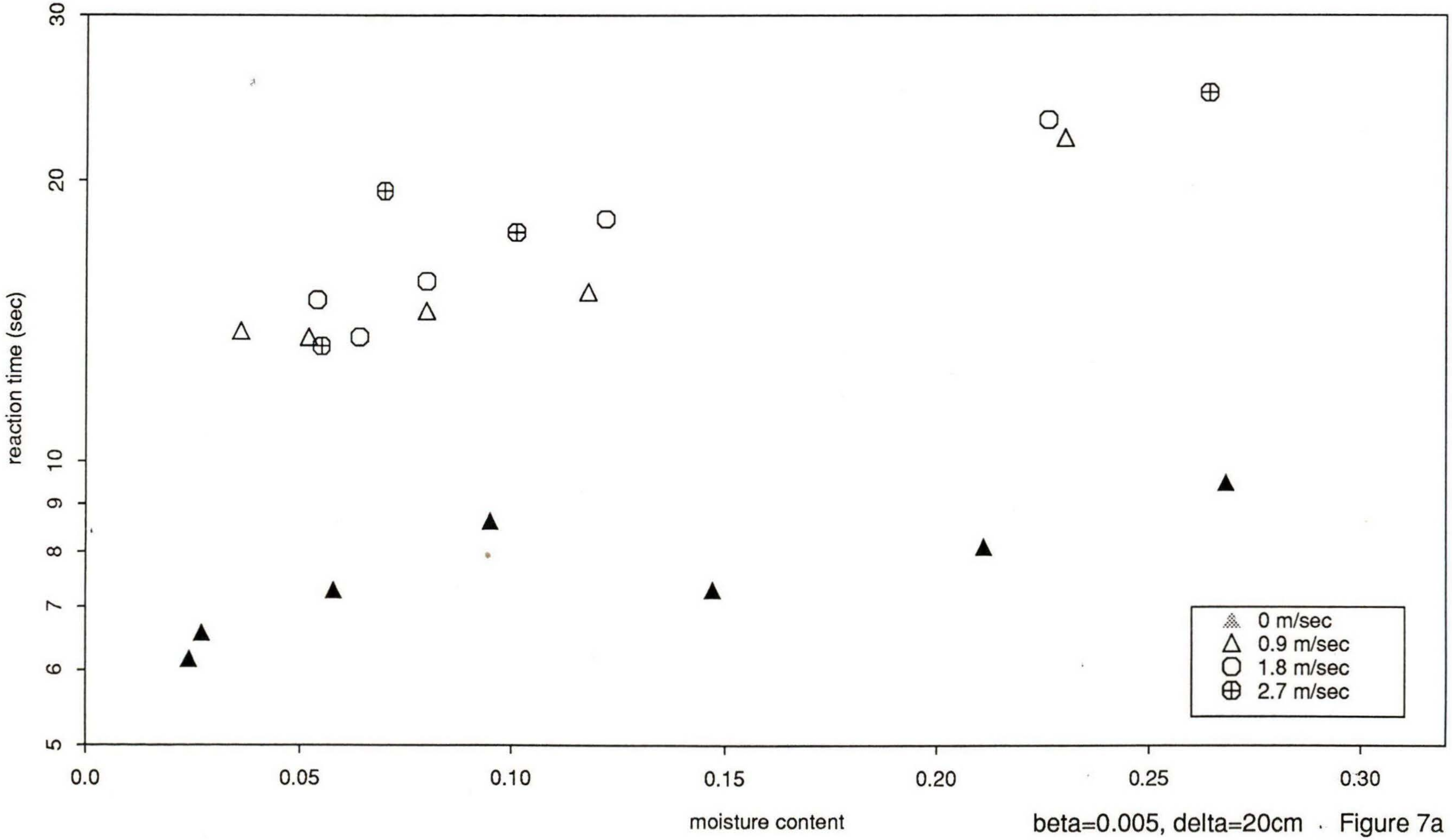


Figure 7

Excelsior --reaction time versus moisture content



beta=0.005, delta=20cm . Figure 7a

Excelsior - ambient conditions -- Propagating flux

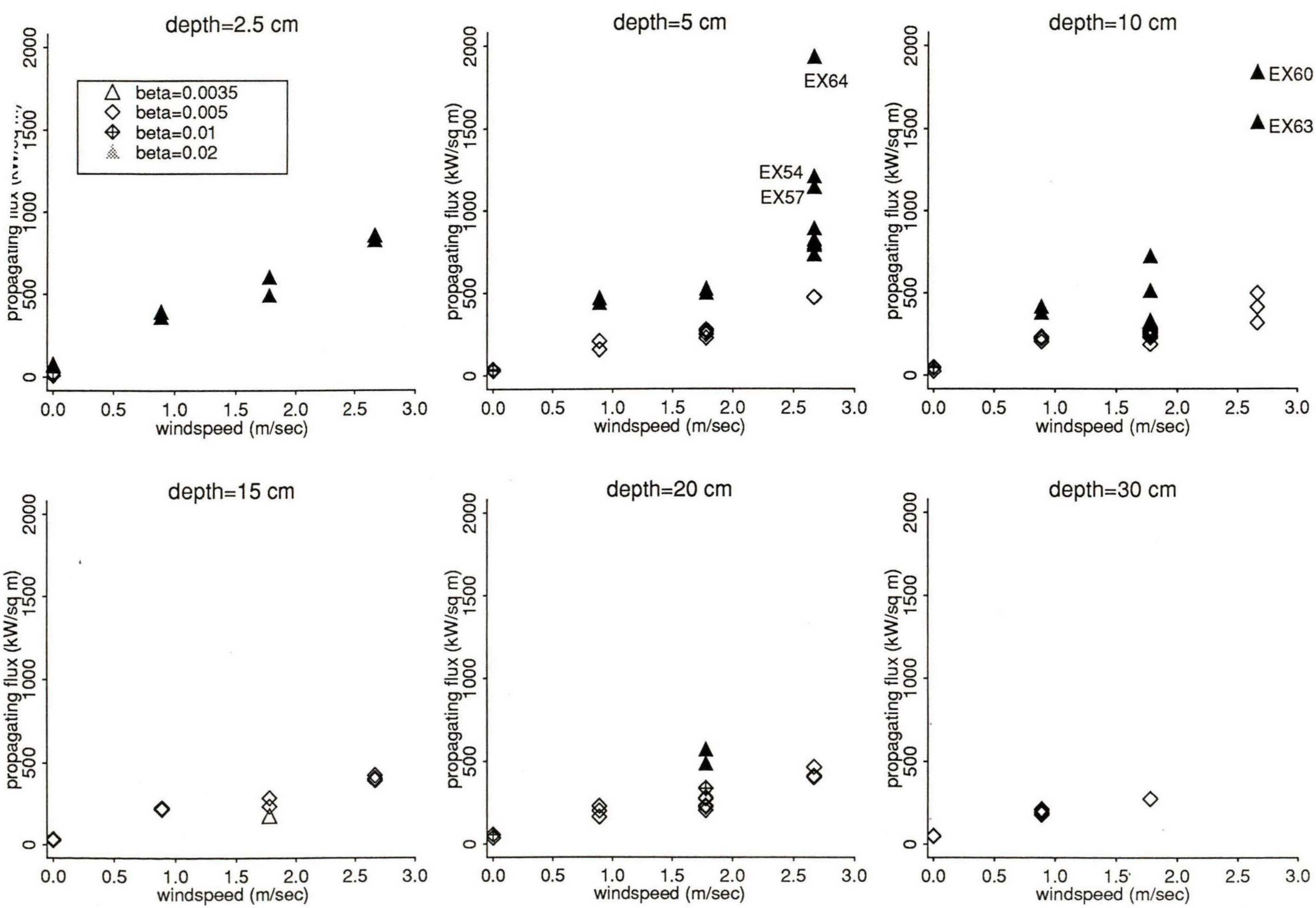


Figure 8

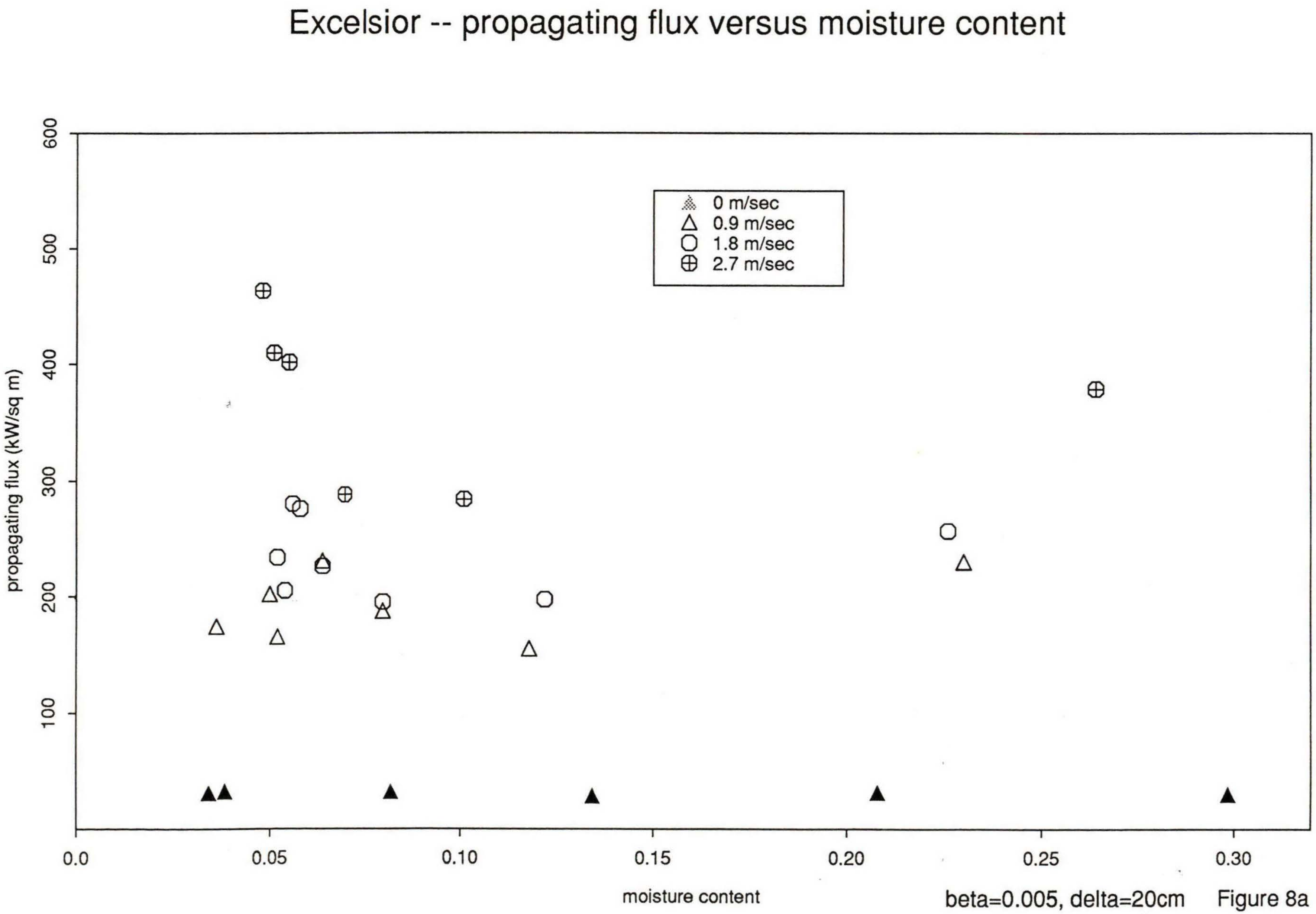

Excelsior -- propagating flux versus moisture content
0 m/sec
0.9 m/sec
1.8 m/sec
2.7 m/sec
propagating flux (kW/sq m)
100
200
300
400
500
600
0.0
0.05
0.10
0.15
0.20
0.25
0.30
moisture content
beta=0.005, delta=20cm

Figure 8a

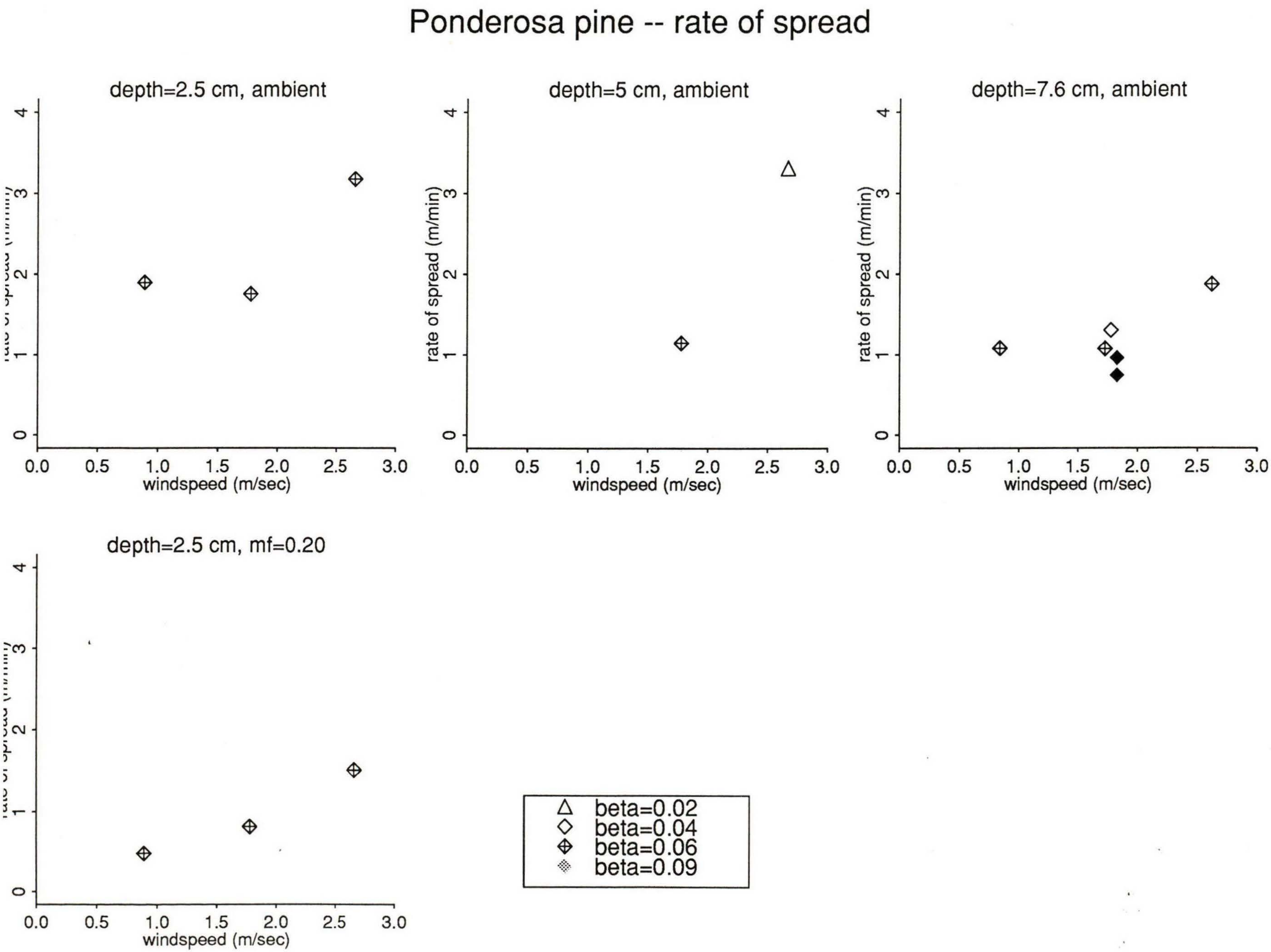

Ponderosa pine -- rate of spread
depth=2.5 cm, ambient
depth=5 cm, ambient
depth=7.6 cm, ambient
depth=2.5 cm, mf=0.20
rate of spread (m/min)
windspeed (m/sec)
0.0 0.5 1.0 1.5 2.0 2.5 3.0
0 1 2 3 4
beta=0.02
beta=0.04
beta=0.06
beta=0.09

Figure 9

Ponderosa pine -- flame length

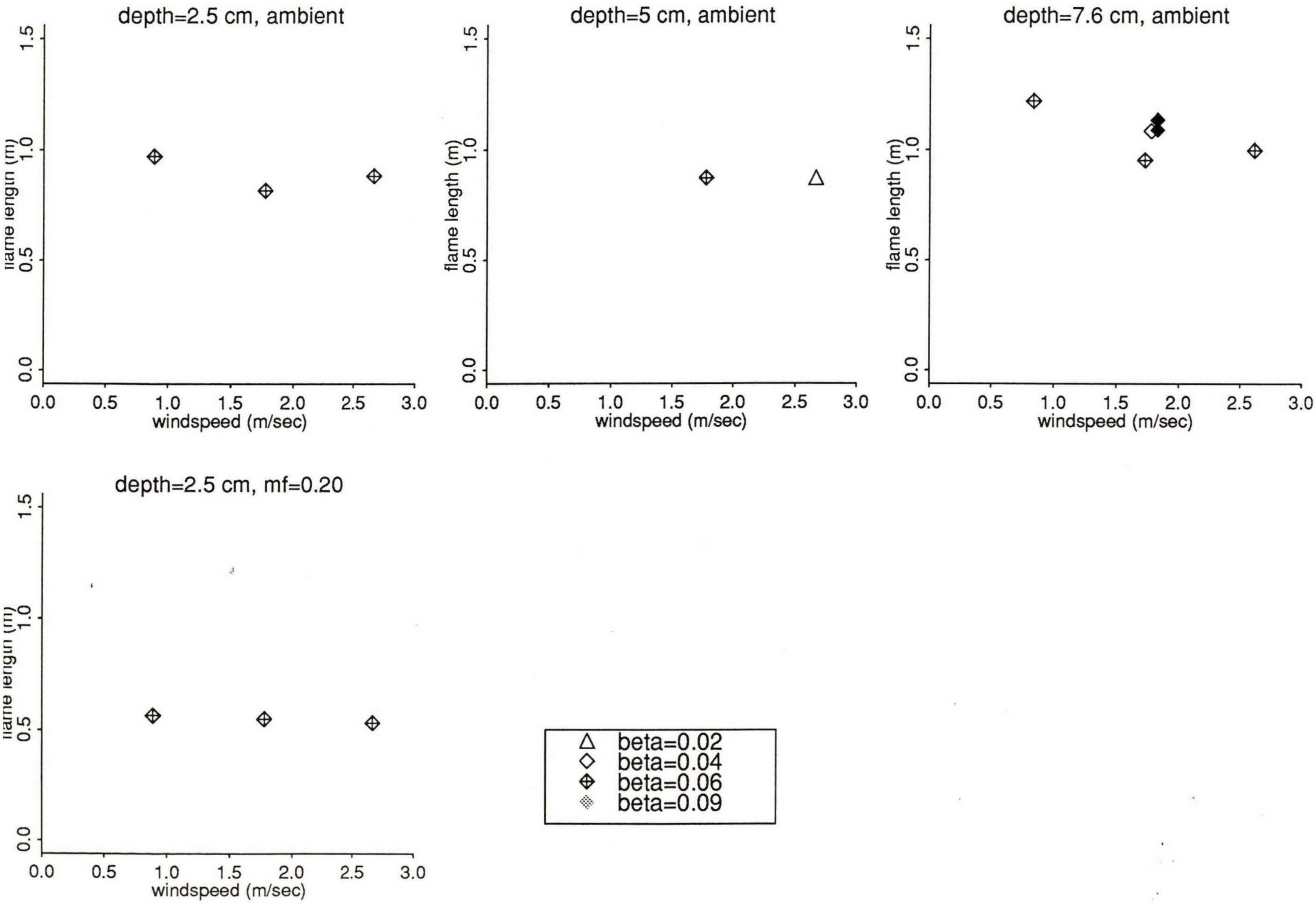


Figure 10

Ponderosa pine -- flame angle

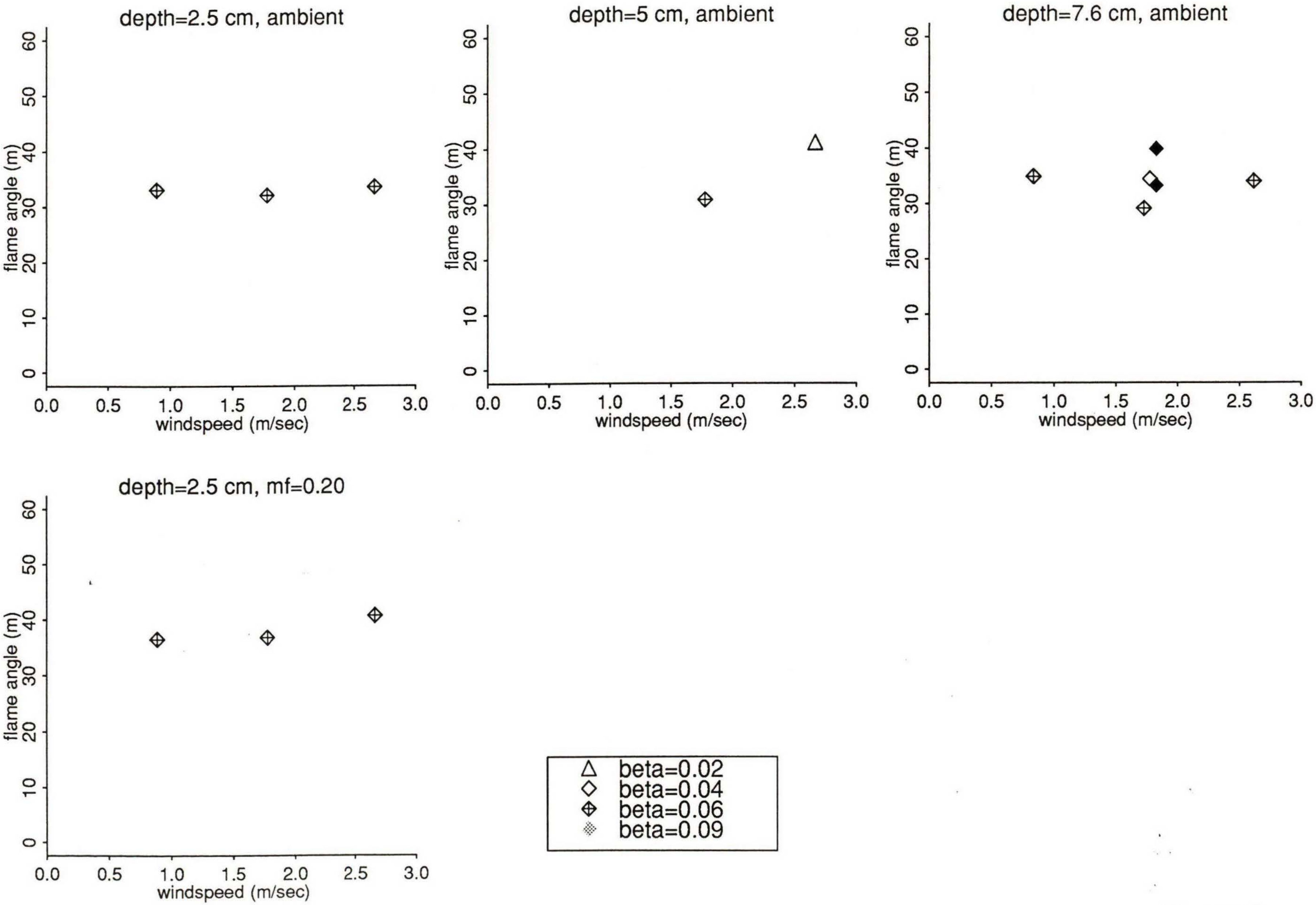


Figure 11

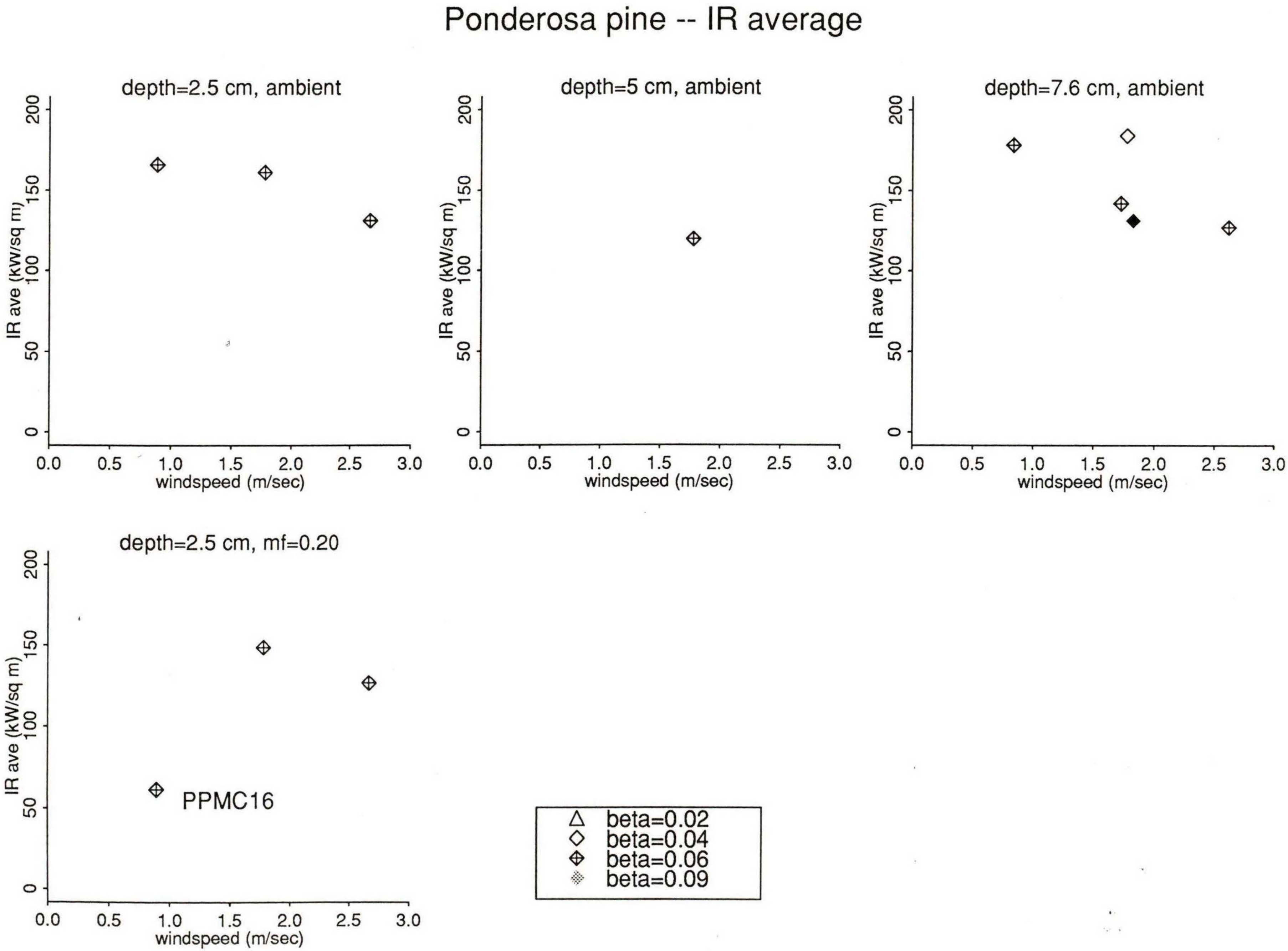

Ponderosa pine -- IR average
depth=2.5 cm, ambient
depth=5 cm, ambient
depth=7.6 cm, ambient
depth=2.5 cm, mf=0.20
IR ave (kW/sq m)
windspeed (m/sec)
0
50
100
150
200
0.0
0.5
1.0
1.5
2.0
2.5
3.0
PPMC16
beta=0.02
beta=0.04
beta=0.06
beta=0.09

Figure 12

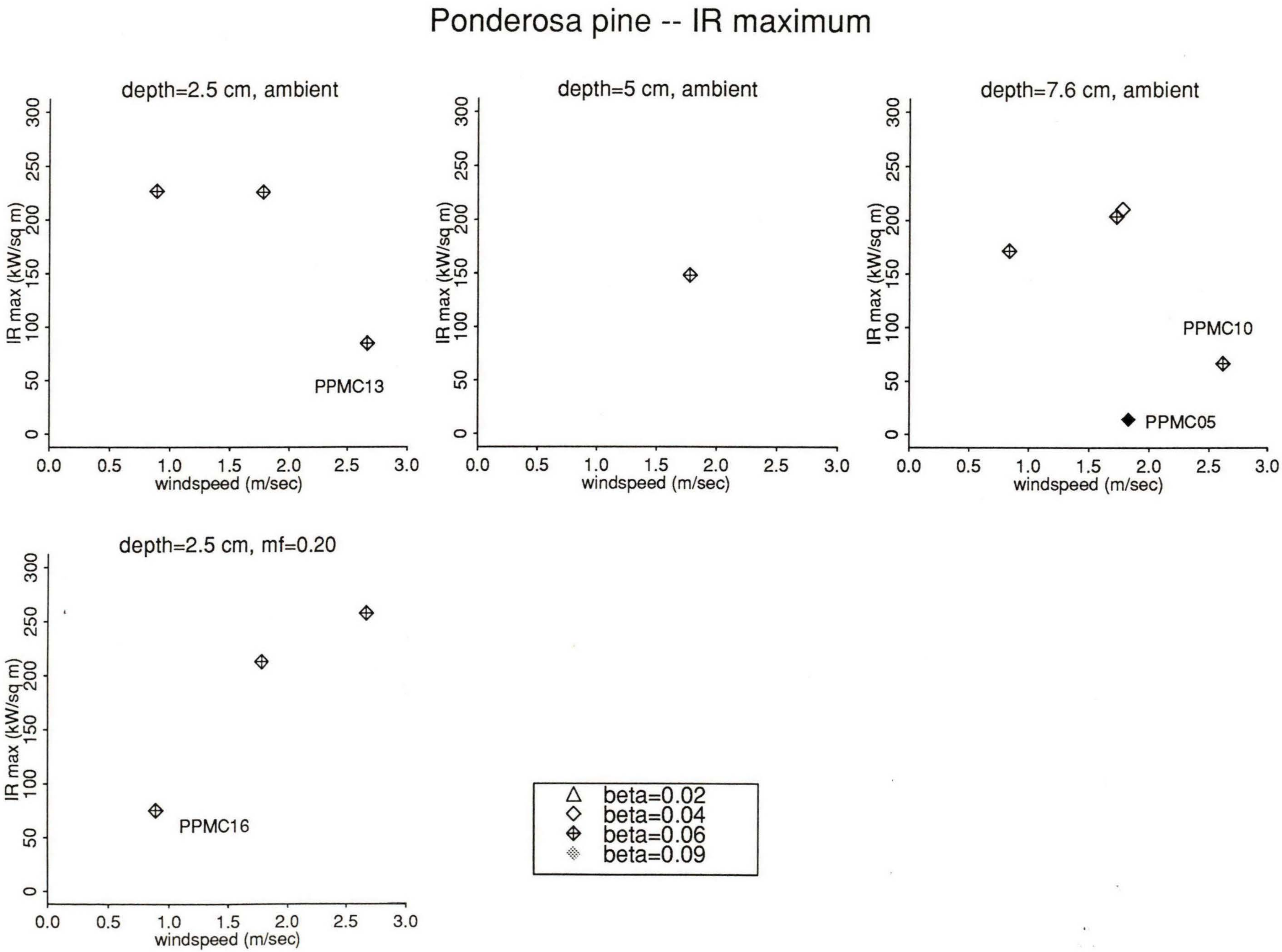

Ponderosa pine -- IR maximum
depth=2.5 cm, ambient
depth=5 cm, ambient
depth=7.6 cm, ambient
depth=2.5 cm, mf=0.20
IR max (kW/sq m)
windspeed (m/sec)
0
50
100
150
200
250
300
0.0
0.5
1.0
1.5
2.0
2.5
3.0
PPMC13
PPMC10
PPMC05
PPMC16
beta=0.02
beta=0.04
beta=0.06
beta=0.09

Figure 13

Ponderosa pine -- Byrams intensity

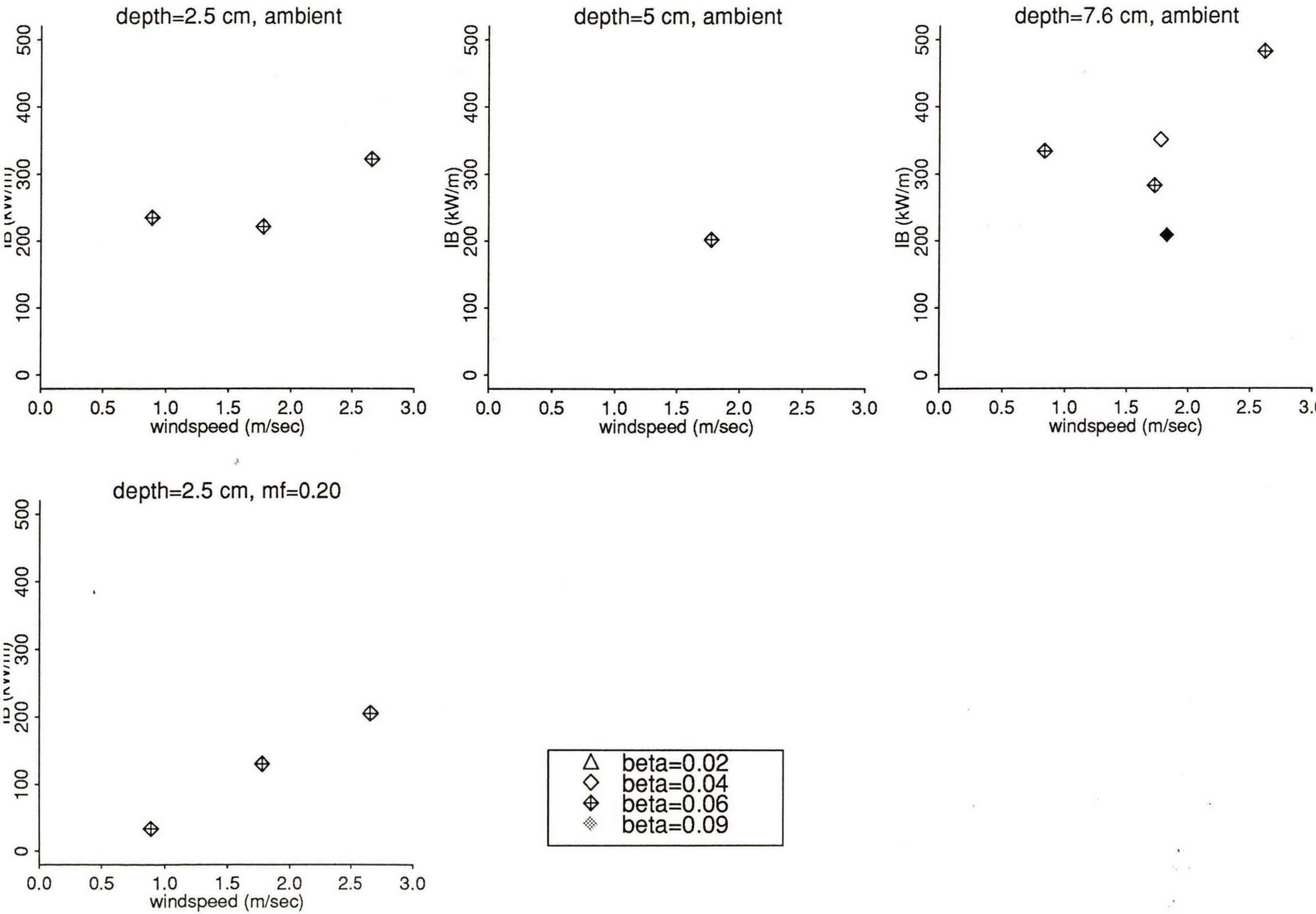


Figure 14

Ponderosa pine -- reaction time

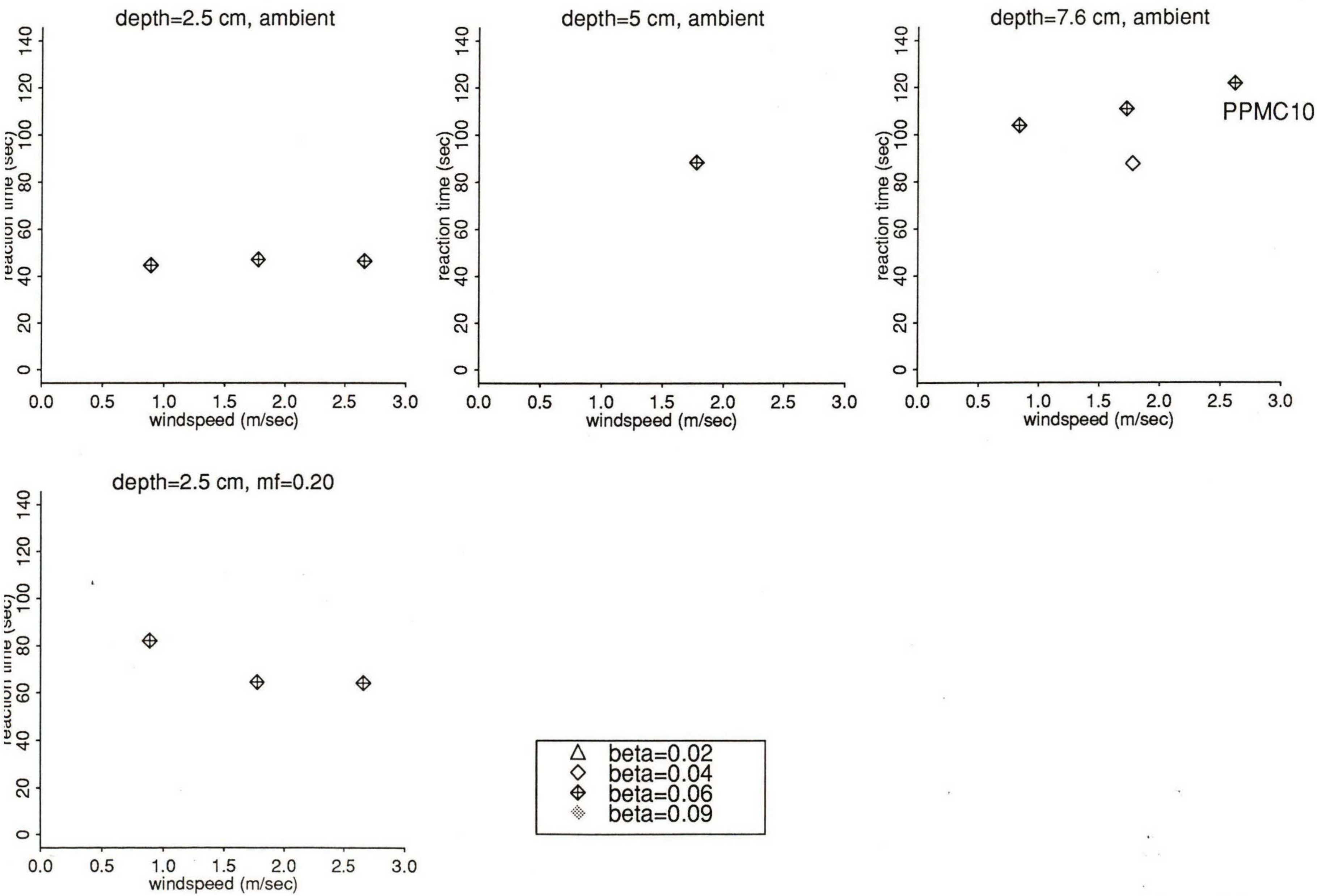


Figure 15

Ponderosa pine -- propagating flux

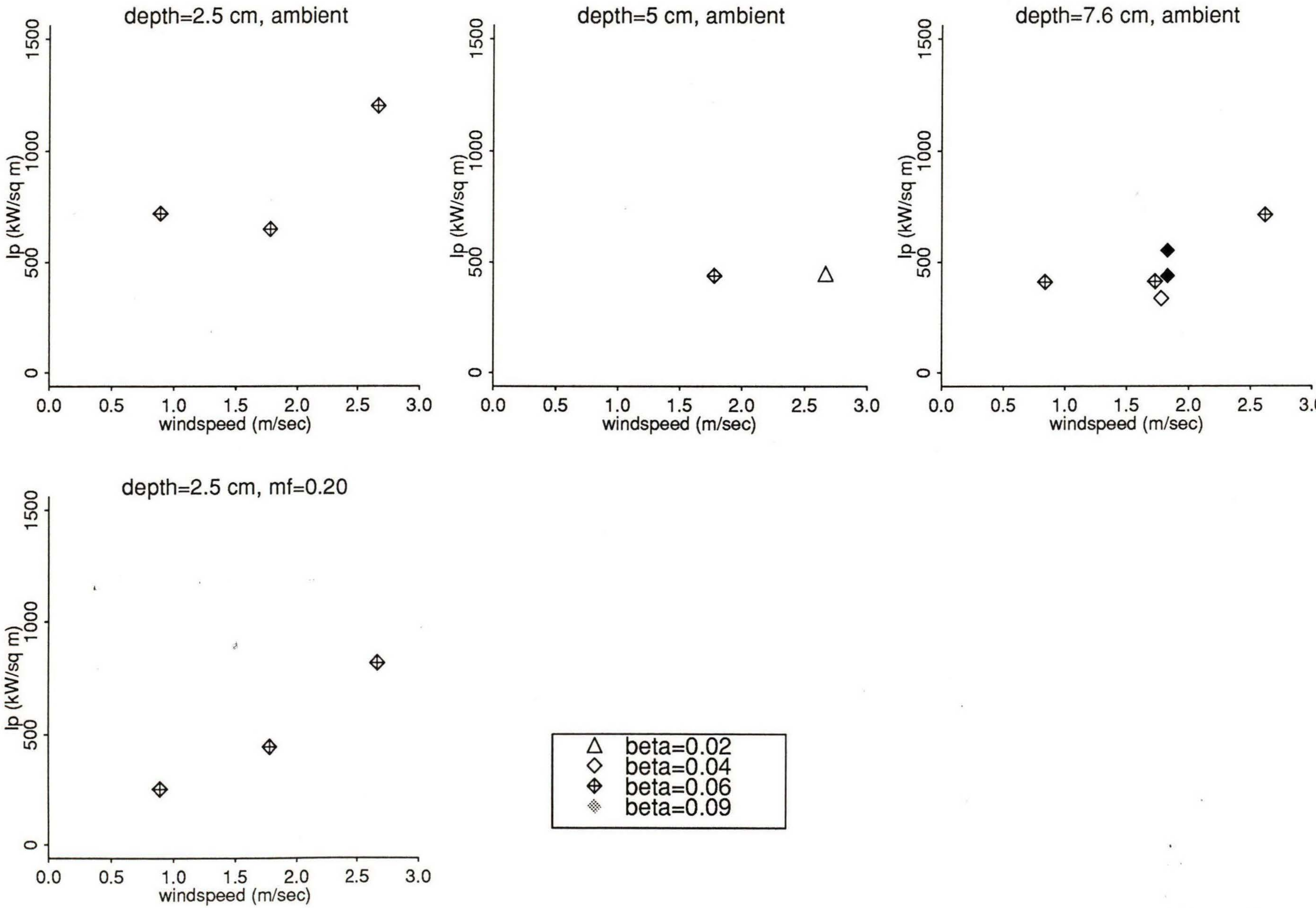


Figure 16

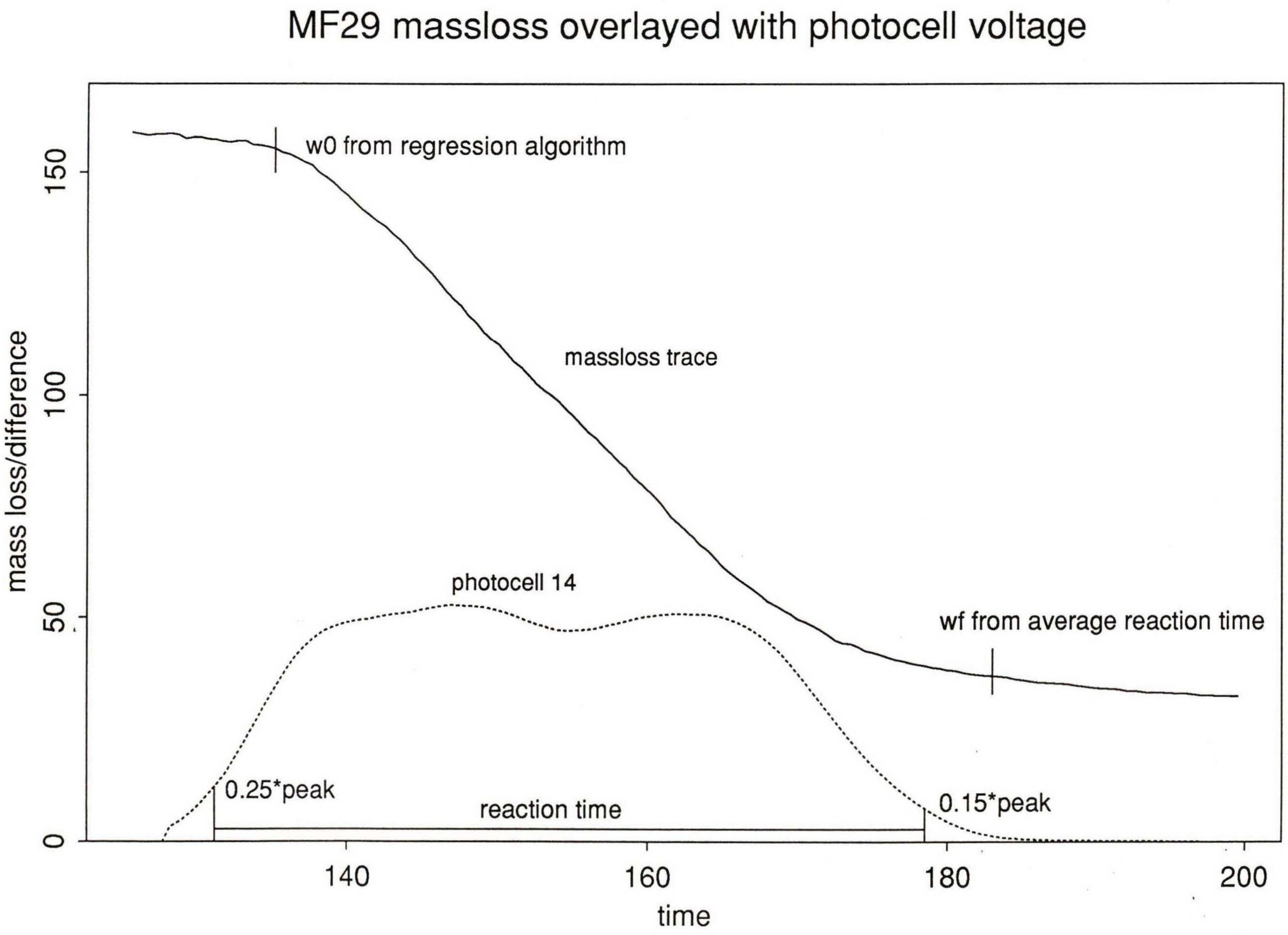

MF29 massloss overlayed with photocell voltage
w0 from regression algorithm
massloss trace
photocell 14
wf from average reaction time
0.25*peak
reaction time
0.15*peak
mass loss/difference
0
50
100
150
140
160
180
200
time

Figure 17

APPENDIX : The Physical Model

1. Methodology

Assume, as in Frandren (1971) & Rothermel (1972) that the fuel absorbs an amount Q of heat (per unit dry mass), after which it spontaneously ignites. The location of points at which ignition occurs is the combustion interface, which is assumed to be travelling at uniform rate R. The equation of motion is based on the conservation of energy of fuel in the pre-ignition phase. No attempt is made to model the chemical combustion process.

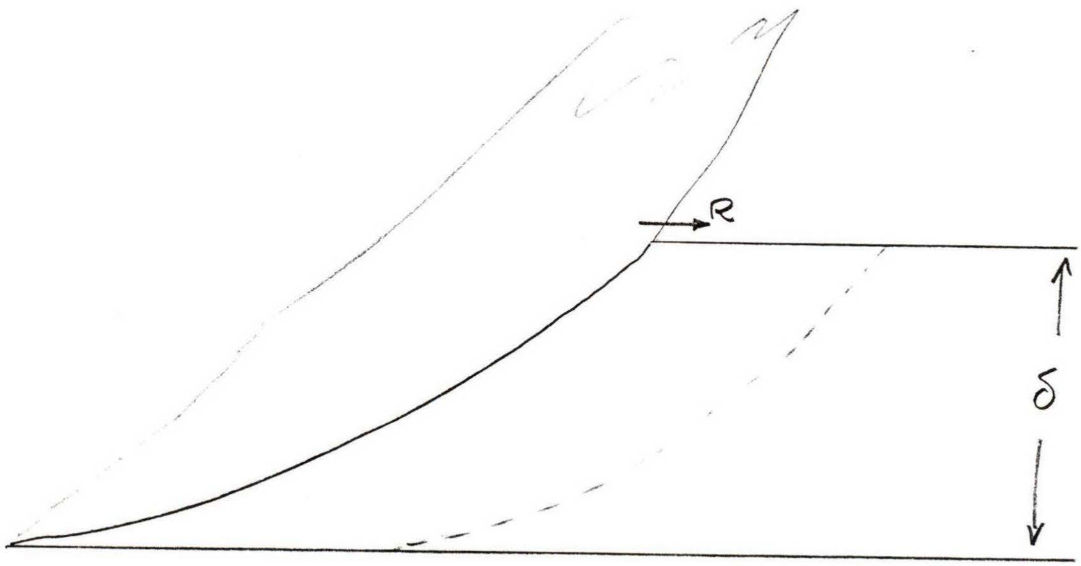


Fig. A1. Combustion interface

Volume swept out by combustion interface in unit time $= R\delta$,
where δ is the fuel bed depth.

$\therefore$ Total (net) rate of heat supply to fuel bed ahead of the combustion interface is

$$R\delta\rho_b\varepsilon Q = I_{\text{tot}} \tag{1}$$

say where ρ_b is the fuel bulk density, and ε is the effective heating number defined in Frandsen (1973). The proposal is to split I_{tot} into contributions from different physical

processes, and to model these separately. Thus we write

$$I_{\text{tot}} = I_{\text{rad}} + I_{\text{diff}} + I_{\text{adv}} - I_{\text{loss}} \tag{2}$$

where I_{rad} is radiative heating, I_{diff} is diffusive heating, caused by local turbulence which results in heat transfer without any mean air flow, I_{adv} is advective heating (convective heating due to a mean air flow), and I_{loss} represents the sum of all heat losses.

In addition each of the heat input terms is split into a contribution from the combustion zone (c) and from the flame (f). Thus for example

$$I_{\text{rad}} = I_{\text{r,c}} + I_{\text{r,f}} \tag{3}$$

Each of the terms in (2) will be modelled by physically-based arguments, but will include constants which need to be determined empirically. Some special experiments will need to be performed to isolate the terms involved. The model will then be fitted to data sets to estimate the remaining parameters. These data sets will include those of Wilson (1990), the current set of experiments underway at IFSL, and field data from the literature.

The modelling of each term will be based on modelling the dependence on

(i) ambient windspeed U.

(ii) fuel geometry parameters σ, β and δ.

(iii) fuel moisture content m_f. The dependence on windspeed and fuel geometry will be considered separately for each heat input term, but the moisture term will probably be dealt with globally.

2. Effect of moisture content

For homogeneous fuels the moisture damping will be modelled as in Wilson (1990): that is

$$I_{\text{tot}} = I_0 \exp\left(-k m_f\right) \tag{4}$$

where I_0 is the total heat input in dry fuel, and k is a constant which depends on σ (and possibly on β and other properties such as presence of waxes and bark covering) (Wilson 1990). For nonhomogeneous fuels we will modify the heat sink term $\delta \rho_b \varepsilon Q$, and the

moisture damping term (4), using ideas from Frandsen (1973) and from Catchpole & Catchpole (1991).

3. Radiant heat inputs

3.1 Radiant heating from the combustion interface.

We model the combustion interface as a plane inclined at an angle θ_c to the vertical, radiating as a black body at temperature T_c. Although this is not entirely realistic (Albini 1985) we do not expect its use to lead to serious error.

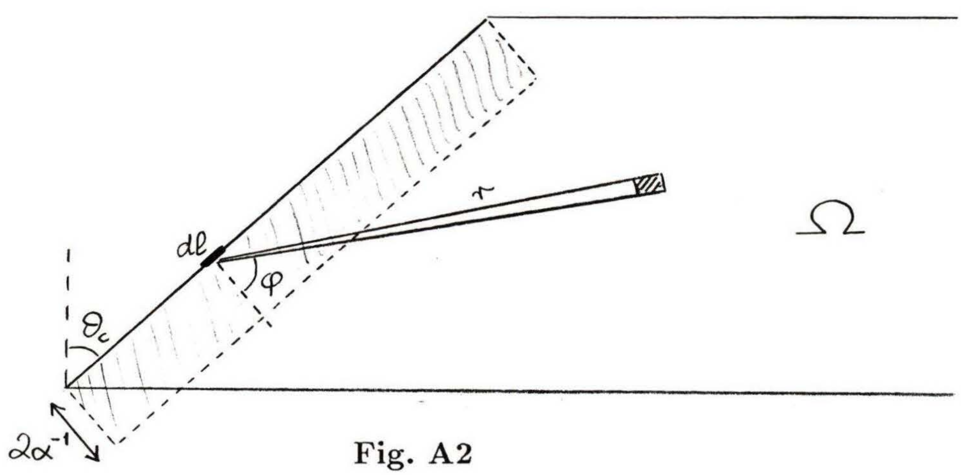


Fig. A2

We model the unburnt fuel as a semi-transparent medium with mean free path length to radiation given by α^{-1}, where

$$\alpha = \sigma\beta/4$$

(Fang 1969). We use a 2-dimensional model and consider the radiation heat emitted per unit time by an element (shown in Fig. A2) of the unburnt region Ω. The radiation emitted by $d\ell$ in the arc $(\varphi, \varphi + d\varphi)$ is

$$\sigma_B \, T_c^4 \, d\ell \, \cos\varphi \, d\varphi.$$

where σ_B is the Stefan-Bolzmann constant, and T_c is the temperature at the combustion interface. This has decayed by a factor $e^{-\alpha r}$ before reaching the area element, and a

proportion αdr is then absorbed. The radiative heat absorbed is thus

$$\alpha\sigma_B T_c^4 \cos\phi \, e^{-\alpha r} dr\, d\phi\, d\ell \, .$$

The total heat absorbed per unit time by the fuel bed is thus

$$I_{\mathrm{r,c}} = \alpha\sigma_B T_c^4 \int d\ell \int \cos\varphi \, e^{-\alpha r} \, dr \, d\varphi \tag{5}$$

(cf. Albini 1985) It may be possible to evaluate (5) analytically, but for the moment a crude approximation will suffice. In order to estimate the proportion of radiation heat lost through failing to be intercepted by a fuel particle (most of which we presume will be absorbed by the ground), we suppose that radiation travels perpendicular to the combustion interface, and is absorbed uniformly in the range $(0, 2\alpha^{-1})$, so that its mean path length is still α^{-1}. The proportion of radiation lost is then the ratio of the shaded triangle to the rectangle shown in Fig. A2, that is

$$\frac{\frac{1}{2} \cdot 2\alpha^{-1} \cdot 2\alpha^{-1} \tan\theta_c}{2\alpha^{-1} \, . \, \delta \sec\theta_c} = \frac{\sin\theta_c}{\alpha\,\delta} \tag{6}$$

Thus as a first approximation,

$$I_{\mathrm{r,c}} \approx \sigma_B T_c^4 \cdot \delta \sec\theta_c \cdot \left(1 - \frac{\sin\theta_c}{\alpha\delta}\right), \tag{7}$$

since $\delta \sec\theta_c$ is the length of the radiating zone.

As noted above, (7) may be replaced by a better approximation in due course. But (7) does illustrate the importance of the various parameters, in particular the dependence on the unknown interface slope θ_c. The combustion interface temperature T_c will be assumed to depend on the fuel moisture in some undetermined manner. This will allow the incorporation of a moisture damping term.

Conclusion: $I_{\mathrm{r,c}}$ can be modelled adequately by (7) but some experimentation may be required to model the combustion interface (average) slope θ_c. The mass loss traces may be useful here.

4. Convective heat inputs

4.1. Advective heating

This relies on there being a mean flow of hot gas through the uncombusted fuel. It is probably a good enough approximation to say that this flow will exist only if the windspeed is nonzero, so that this term will not appear in the zero wind model. 4.1.1. Advective heating from the combustion zone

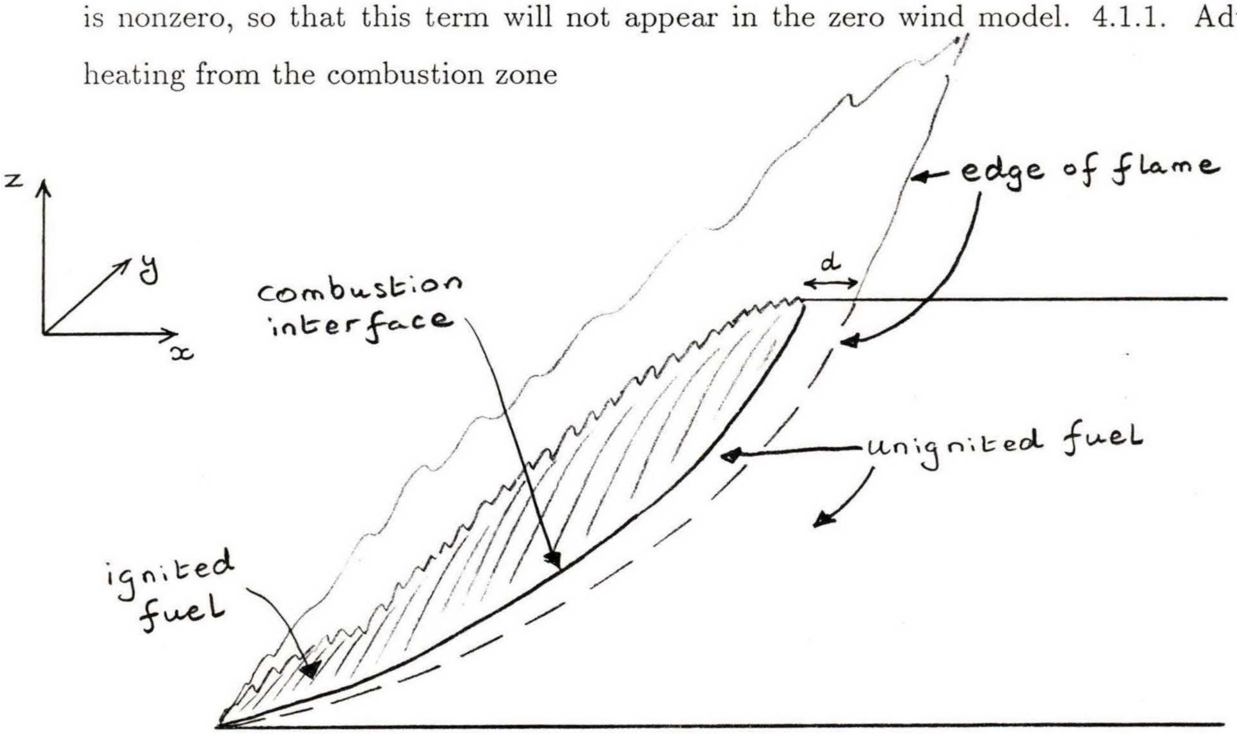


Fig. A3

This figure is similar to one shown in Baines (1990). The distance ℓ by which the flame front leads the combustion interface at the fuel surface will be taken as characterising $I_{a,c}$. Clearly ℓ will increase with windspeed, U. Also visual observation of laboratory fires gives the impression that, for a given windspeed, ℓ is much larger for $\frac{1}{4}''$ sticks than for excelsior.

We assume that a (possibly intermittent) wind, of average strength U_c, penetrates the unburnt fuel to an effective distance of ℓ ahead of the combustion interface, and that this wind is at flame temperature T_f. The surface area of fuel particles, in a volume of dimensions (ℓ, 1, δ) in the (x, y, z) directions, is $\sigma\beta\delta\ell$. For excelsior (diameter $D \approx 0.005$m) in a wind of 1m/sec at 600K the Reynolds number is

$$R_e = \frac{UD}{\nu} \approx \frac{1 * 0.005}{50 * 10^{-6}} = 100 \qquad (8)$$

(Incropera & de Witt), and so the appropriate correlation for forced convective heating is (Incropera & de Witt eqn 7.29)

$$N_u \quad = \quad \frac{\bar{h}D}{k} \quad \approx \quad 0.683 \; R_e^{0.466} P_r^{\frac{1}{3}} \tag{9}$$

The rate of heat transfer is thus

$$\begin{aligned} I_{a,c} &= \bar{h} \; * \; \text{temp. difference} \; * \; \text{surface area} \\ &= \frac{k}{D} \, 0.683 \, (\frac{U_c D}{\nu})^{0.466} \;\; P_r^{\frac{1}{3}} (T_f \; - \; T_{ig}) \, \sigma \beta \delta \ell \\ &= \frac{k\sigma}{4} \, 0.683 \, (\frac{4U_c}{\sigma\nu})^{0.466} \;\; P_r^{\frac{1}{3}} (T_f - T_{ig}) \, \sigma \beta \delta \ell \\ &= \frac{0.326 \; k \; P_r^{\frac{1}{3}}}{\nu^{0.466}} \;\; U_c^{0.466} \; \sigma^{1.534} \; (T_f - T_{ig}) \; \beta \delta \ell \end{aligned} \tag{10}$$

where P_r = Prandtl number

k = thermal conductivity

ν = viscosity.

The unknowns in (10) are

U_c = effective mean wind velocity through combustion interface

ℓ = effective penetration distance of wind into unburnt fuel.

It is quite easy to conceive of an experimental set-up to measure ℓ (which might depend on U and $\sigma\beta$ for example) by putting an air thermocouple (to measure time of arrival of flame) and a thermocouple inside a fuel element (to measure time of ignition) into experimental fuel beds. It is very difficult to measure U_c however. Since U_c and ℓ are likely to depend on the same things we may have to resort to empirical relations of the form

$$\ell \; U_c^{0.4486} \quad = \quad \begin{cases} 0, & U < U_0 \\ f(\sigma\beta) \; g(U - U_0), & U > U_0 \end{cases} \tag{11}$$

which can be determined only by comparing with rates of spread from experimental fires. Here U_0 is a "threshold wind speed", which may depend on the power of the fire.

Conclusion

$$I_{a,c} \;\; \approx \;\; \sigma^{\frac{3}{2}} \beta\delta \;\; f(\sigma\beta) \; g(U - U_0) \tag{12}$$

where f, g and U_0 need to be empirically determined by comparing predictions of rate of spread with the results of experimental fires.

4.1.2 Advective heating from the flame.

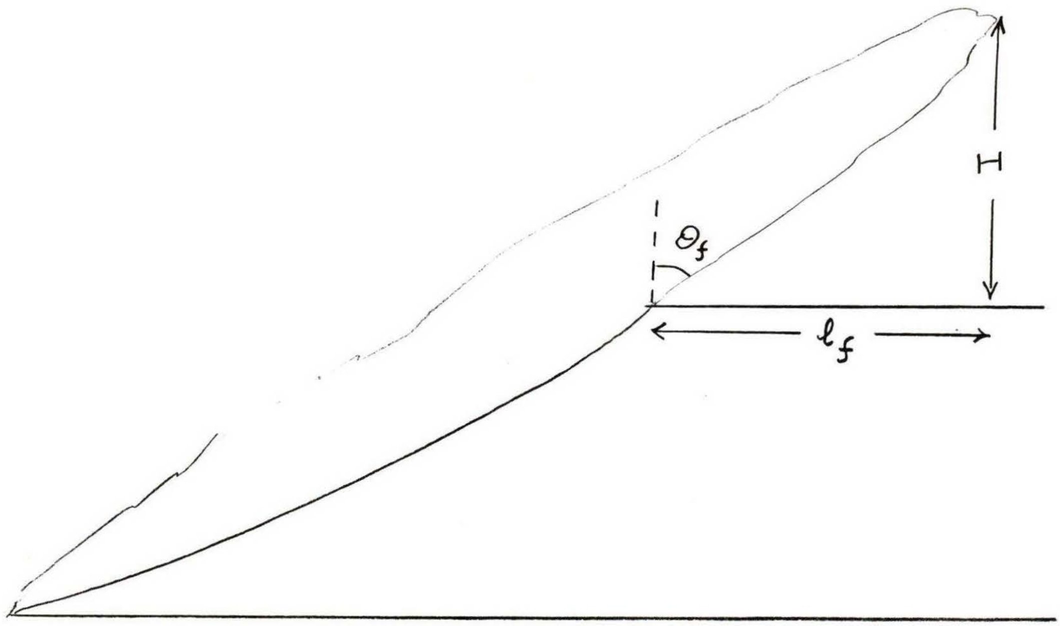


Fig. A4

Intermittent heating due to flame "licking" onto and through the fuel bed. Distance of hot gas (ℓ_f) may be observable from videos, and we might observe for example

$$\ell_f \approx H \tan \theta_f \quad .(13)$$

The average depth λ to which this flame penetrates the bed can be measured by putting air thermocouples into the fuel bed. The air speed U_s may be taken as the wind speed along the fuel bed surface. The heat transfer would then by given by a term similar to (10):

$$I_{a,f} \approx \frac{0.366\ k\ P_r^{\frac{1}{3}}}{\nu^{0.466}}\ U_s^{0.466}\ \sigma^{\frac{3}{2}}\ \beta\lambda\ell_f \quad (14)$$

Conclusion We can use (14) empirically, with U_s, ℓ_f and λ determined so as to match rate of spread from test fires. If we do this then we might well lump $I_{a,c}$ and $I_{a,f}$ together. Or we can perform experiments to determine U_s, ℓ_f and λ.

Remarks on convective heat transfer

We have assumed, for calculating Reynolds numbers and in the choice of appropriate empirical heat transfer equation, that we are dealing with the laminar flow of air over a single cylindrical particle oriented perpendicular to the flow. In fact

(a) the particles may be oriented at random (e.g. excelsior)

(b) the particles may be so close together that they behave more like a porous bed than isolated particles (e.g., pine needle litter)

(c) the flow may be turbulent rather than laminar.

The effect of each of these needs to be estimated.